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VOL. 1. No. 6 NOVEMBER 1940

THE ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

PRINCIPAL CONTENTS

Engineering for the Higher Altitude.

Oxygen Boosting of Airplane Diesel Engines.

New German Machine Tools.

Marine Gas Turbines.

"D.V.L." Accelerometer.

Axial Vibration in Crankshafts.

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Published by Engineers' Digest Ltd., 59, Weymouth Street, London, W.1
Distribution: E. & F. N. Spon, Ltd., 57, Haymarket, London, S.W.1
Price £2 2 0 per year

THE ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

Published Monthly at 58, Weymouth Street, London, W.1

Publishers: **ENGINEERS' DIGEST LTD.** - - - Telephone: WELBECK 9357

Advertisement Dept. - 21, Warwick Lane, London, E.C.4
Telephone: City 6071. Telegrams: Adanimate, Cent, London

Distribution - - E. & F. N. SPON, LTD., 57, Haymarket, London, S.W.1
Telephone: Whitehall 1860. Telegrams: Fenspon, Lesquare, London

Editorial Dept. - - 59, Weymouth Street, London, W.1
Telephone: Welbeck 9357

Subscriptions to: ENGINEERS' DIGEST LTD., 59, Weymouth Street, London, W.1, or E. & F. N. SPON, LTD., 57, Haymarket, London, S.W.1

Annual Subscription of 12 numbers, £2 2s. Single Copies, 4/6

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CONTENTS

AERONAUTICS: *page*

D.V.L. Accelerometer 227
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 227.

Engineering for the Higher Altitude 196
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 196.

Estimating the Efficiency of Light A.A. Guns, with respect to their Sighting Device 199
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 199.

Drop-Test of Shock Absorber 201
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 201.

"LETOV" Drop Test Machine 203
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 203.

Oxygen Boosting of Airplane Diesel Engines 217
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 217.

ELECTRICAL ENGINEERING:

Air-Blast Circuit Breakers as a means of dealing with Short-Circuits by Auto-Closing 220
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 220.

Failures in Overhead Lines with Aluminium Conductors 215
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 215.

Portable Brightness Meter for Luminous Preparations 208
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 208.

Telephone Cable Suspender 214
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 214.

HEATING AND VENTILATING:

Processing with Dowtherm 222
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 222.

MECHANICAL ENGINEERING:

All-Angle Milling Machine 227
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 227.

Axial Vibration in Crankshafts 206
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 206.

German Machine Tool Accessories 229
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 229.

"Low-Head" Cooling Tower 228
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 228.

Lubrication and its Relation to Maintenance 205
"*The Engineers' Digest*," *Vol.* 1 *No.* 6, *p.* 205.

Marine Gas Turbines 204
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 204.

New German Machine Tools 191
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 191.

Present Methods of Turning Camshafts 209
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 209.

METALLURGY:

Application of Demagnetising and IR Drop Compensation to Cold Strip Mills 225
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 225.

Factors Affecting Production of Blast Furnace Gas 224
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 224.

German Machine Tool Accessories 229
"*The Engineers' Digest*," *Vol.* 1, *No.* 6, *p.* 229.

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NEW GERMAN MACHINE TOOLS

(From *Die Werkzeugmaschine*, Berlin, Vol. 44, No. 5, March, 1940, pp. 87-118).

(*Continued from page* 160).

THE starting and stopping of the work piece holding spindle is controlled by the long levers on the upper part of the machine. An adjustable overload device stops the machine when the pressure on the boring tool is excessive. The feed gear is automatic and limited by an adjustable stop. The feed can also be stopped by hand at any point. The automatic feed can be adjusted to four different speeds.

Planing and slotting machines.

Fig. 20 shows a new type of hydraulically operated heavy duty slotting machine manufactured by the Gustloff works. The machine is of sturdy and self-contained construction. The speed of the ram can be steplessly adjusted from 4 to 40 m./min. 12 different feeds from 0.2 mm. to 2.4 mm. are available for longitudinal, circular and transverse movement of the table. The drive is provided by a flanged motor driving a pump, which forces the oil into the cylinders of the ram through the control valves. The ram speed is constant during the stroke, and can be adjusted while the machine is working. The cutting pressure and speed is indicated on a scale. The ram slides in long guides and can be tilted to 30°. Quick travel of the table can be obtained in all directions.

Due to the hydraulic control, elastic cutting is obtained, hence a higher efficiency and longer life of the tools. An automatic tool relieving device is built into the ram. The motor is controlled by push-buttons from the control station.

Fig. 21 shows a special slotting machine, the ram of which can be tilted to 45° at both sides. The table can slide in longitudinal, circular and transverse directions on the wide bed. The column is attached to the back of the bed. A Leonard motor of about 70 H.P. provides the driving power for a maximum cutting power of 8000 kg. adjustable in the ratio 1:6 according to the cutting speed.

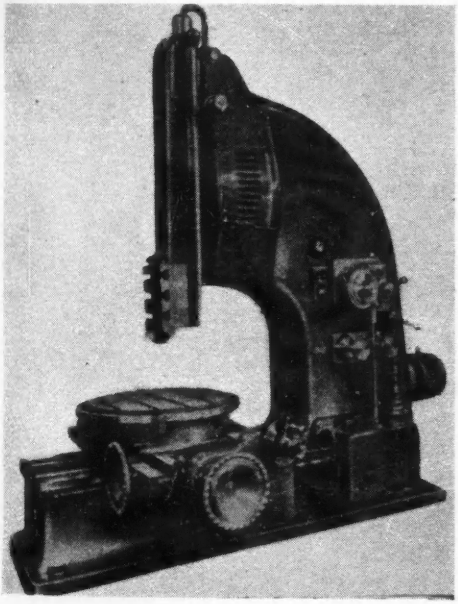


Fig. 20.

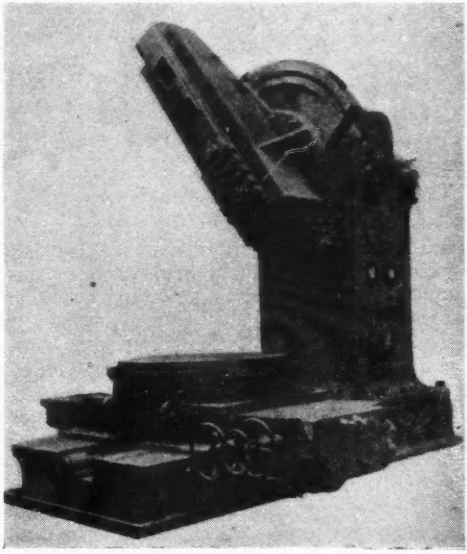


Fig. 21.

Fig. 22 shows a special portal slotting machine for large and heavy work pieces. The cutting speed can be adjusted to 4-16 m. per minute, and the return speed is 20 m./min. In spite of the comparatively high cutting and return speed one spindle serves to move the ram. The table can only move in circular and longitudinal directions, the transverse feed being adjusted on the ram. A 100 H.P. Leonard motor drives the machine, which has a maximum cutting pressure of 12,000 kg.

Grinding machines.

The surface grinding machines DWH 900 and 1200 (Fig. 23), are fitted with Diskus-segmental grinding wheels. The latter are carefully protected against dust particles. The grinding shaft runs in carefully designed roller

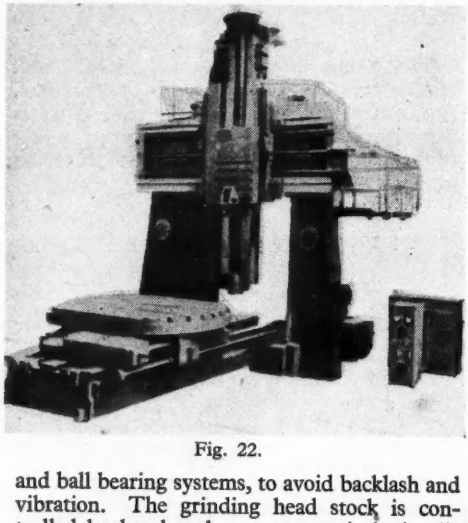


Fig. 22.

and ball bearing systems, to avoid backlash and vibration. The grinding head stock is controlled by hand or by an automatic hydraulic device.

The Diskus segment grinding wheel consists of a supporting disc with a steel jig screwed on. The latter carries the grinding segments, allowing for a continuous wearing off and a quick changing of the wheels. The table is hydraulically operated, and this accounts for the shockless movement of the table, quick adjustability of the feed, and the appropriate travel of the work piece. The speed regulation is stepless.

The column of the circular table surface grinding machine (Fig. 24), is made in one

Fig. 23.

piece with the bed. The Diskus segment wheel with its shaft and bearings is carried on the vertical slide, while the 26 H.P. motor is fixed on the front of the column. The base of the bed serves as a water container for wet grinding. The table is ring-shaped with 1,000 mm. outside dia. and 350 mm. inside dia. The speed of the table can be adjusted step-

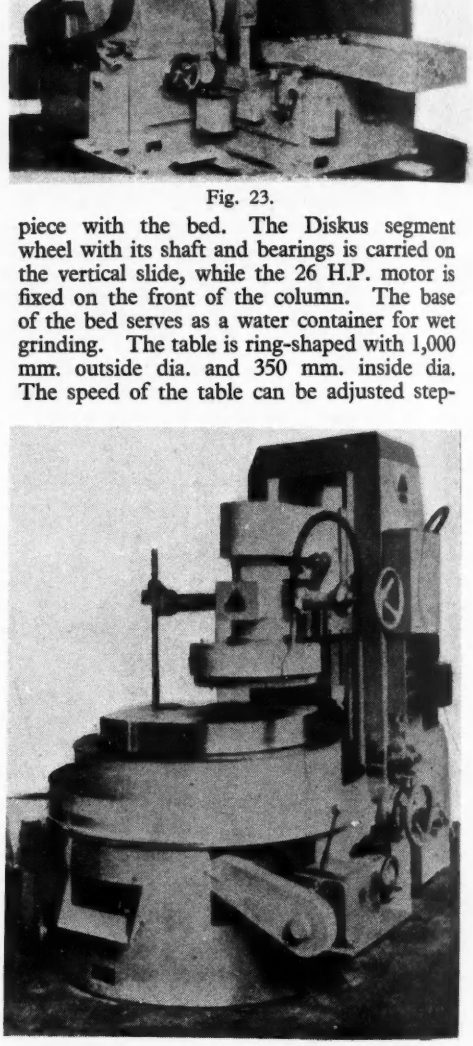


Fig. 24.

lessly. The grinding head is provided with a fast travel arrangement for cleaning, gauging and reloading. A fine micrometer adjustment allows for a regulation of the order of 0.005 mm. This adjustment is hydraulically or hand operated.

Fig. 25 shows a centreless grinding machine BEF 1. Two grinding methods are used with this machine: continuous grinding and plunge cut grinding. With the first method, the work piece is forced between the grinding wheel and the feed wheel, the latter having a small inclination to the vertical plane. Thus, cylindrical work pieces can be ground without collars.

With the second grinding method, work pieces with collar and with set up profiles can be ground. The work piece is kept in position by an end stop which serves also as an ejecting rod. The slide is moved towards the work piece up to a strong stop. Fine adjustment of this feed is available. By moving back the grinding wheel to the original position, the work piece is ejected. This method is also suitable for grinding conical work pieces.

The machine shown in Fig. 25 has additionally an automatic grooving device, which can be arranged so that it operates automatically after being adjusted for the first work piece. In this case, the feed is also automatically controlled, hence, uniform grinding of the work pieces is obtained.

Fig. 26 shows a cylindrical grinding machine for journals and crank shafts as well as for conical work pieces. The work piece is kept in position by a fixed spindle and back centre, and a driving plate provides the feed. The table moves on the bed in V grooves and surface guides. The upper part of the table can be tilted to grind conical work pieces. The angle of inclination can be adjusted on a scale. The feed of the table is hydraulically operated and can be adjusted to 0.4 m./min. The table travel is limited by means of stops.

The work piece spindle is fixed. The driving plate is rotated by a motor with adjustable speed. The motor driving the grinding wheel is fixed on to the head stock. Fast travel and hydraulically operated grooving device is provided. The adjustable rate of feed is indicated on a gauge.

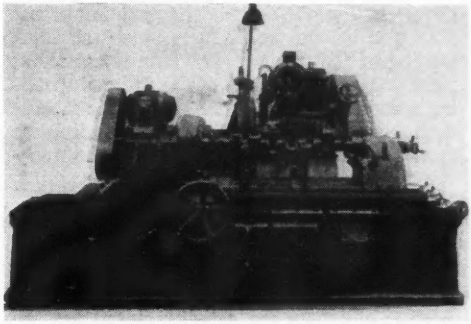


Fig. 26.

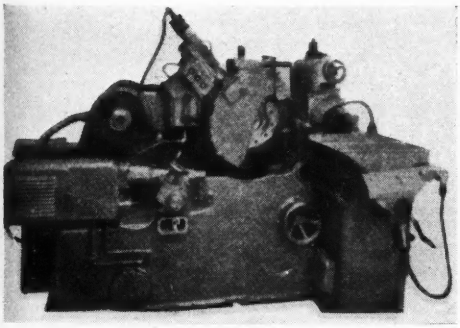


Fig. 25.

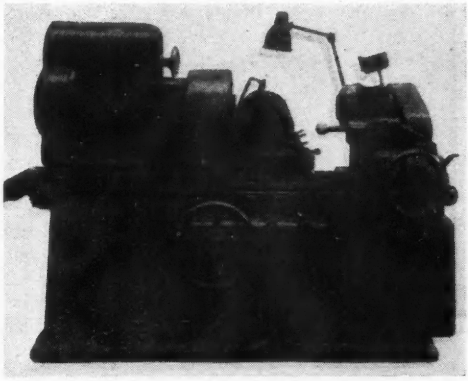


Fig. 27.

An internal and face grinding machine is shown in Fig. 27. Since the grinding of the bore and of the face is done with the same loading of the machine, the bore is exactly at right angles to the face. The internal grinding spindle is fixed and the feed of the table is hydraulically controlled. During grinding the face, the grinding disc is moving in the transverse direction and the work piece is rotating. Fine adjustment of the feed is arranged.

Fig. 28 shows an automatic internal grinding machine. Grinding is carried out in two phases, adjustment of the grinding wheel taking place between the two phases. For this purpose not only the grinding wheel but the slide too, is adjusted so that the conditions for the two phases are identical. Thus, a very accurate grinding is obtained. The grinding spindle is not fixed rigidly to the cross saddle, but is suspended in a pendulum, hence, the feed can be very finely adjusted and is independent of the weight and the cohesive force of the saddle. A steering box is provided in the front wall of the machine to enclose the entire hydraulic apparatus.

Gear wheel cutting machines.

Fig. 29 shows a rolling cutting machine for axle drive bevel wheels only. Universal machines used for cutting pinions as well, cannot be provided with spindles of such large diameter as single purpose machines. Fig. 30 shows the spindle of a single purpose machine. Its approximate weight is 600 kg. Due to this large weight of the spindle, vibration of the machine is reduced, and hence, these type of machines can run at a much higher speed.

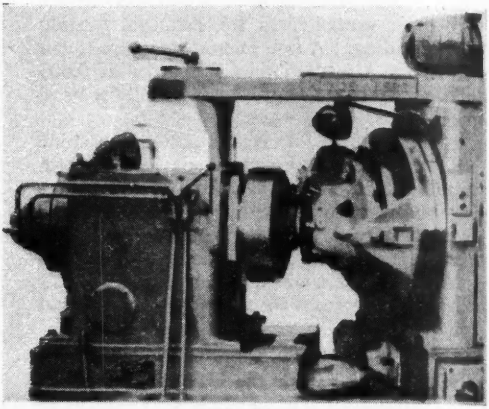


Fig. 29.

Electric drive.

An individual electric driven vertical milling machine is shown in Fig. 31. The machine and all the electric equipments, e.g. generators, motors, instruments, form one compact piece. The machine is fitted with seven motors, three generators and a cooling pump. Fig. 32 shows the arrangement of the electric equipment.

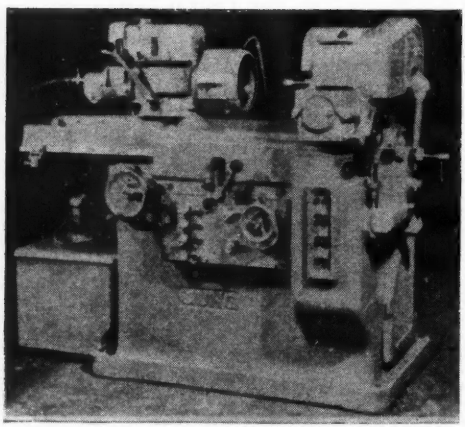


Fig. 28.

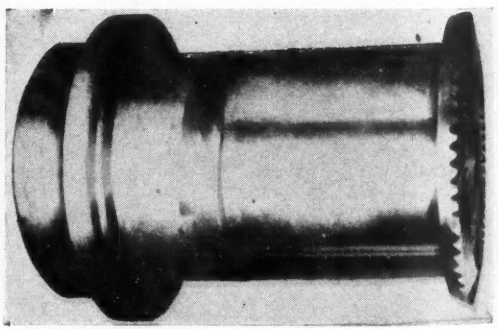


Fig. 30.

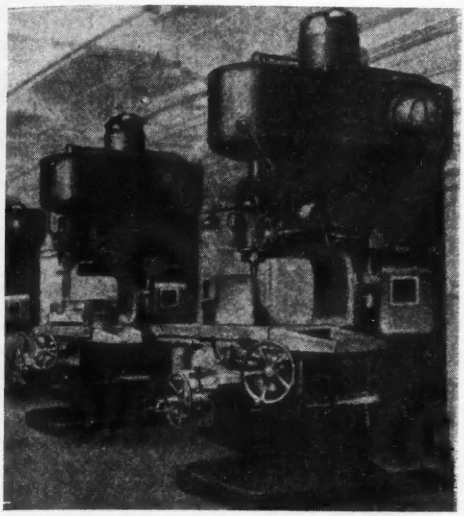


Fig. 31.

(a_1) A.C. driving motor, 15 KW., 2,900 R.P.M. and two generators on the same shaft, 6.5 KW., 230 V. and 4/9.5 KW., 98/220 V. respectively.

(a_2) D.C. generator for the excitation, driven by (a_1).

(b) D.C. motor for the cutter spindle, 2.3/7.5 KW., 300/700/3,000 R.P.M.

(c) D.C. motor for the feed, 7 KW., 600/1,500/3,000 R.P.M.

(d) Cooling pump.

(e) Two A.C. motors for high-speed travel and drive to coupling.

(f_1) and (f_2) Two D.C. motors for regulation.

(g) Electric instruments.

The speed regulation is 1:10 by means of press buttons. The range of feed is 1:125.

Miscellaneous machines.

Fig. 33 shows a circular sawing machine with stepless regulation for the feed drive and the saw cutting drive. Both the feed drive and the saw cutting drive can be adjusted on a scale with logarithmic divisions, i.e. the correct speed can be adjusted according to the shape and material of the work piece. A stepless PIV chain drive is employed. To obtain a smooth and quiet running, a worm gear arrangement is provided. By means of a double gear device, the worm gear is disengaged by 1:7 of the cutting pressure. The feed is hydraulically operated

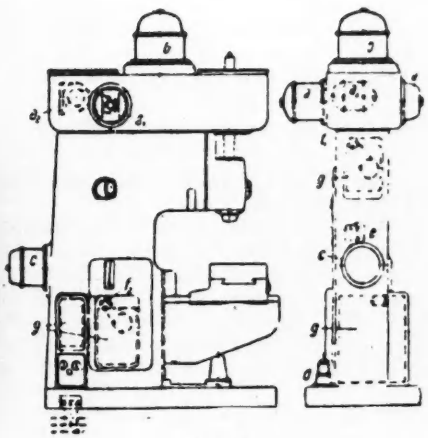


Fig. 32.

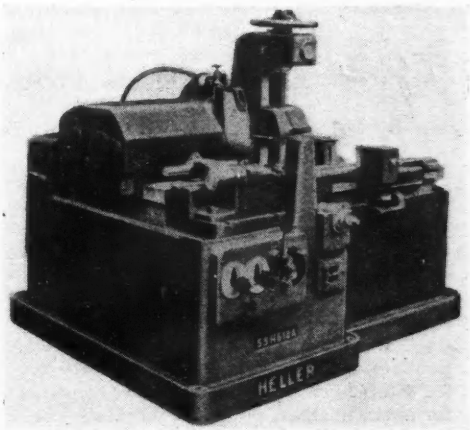


Fig. 33.

by a feed pump and a regulator pump, the latter being controlled by the sawing pressure. Full hydraulic control is used, eliminating cams and couplings. The order of the movements is as follows :

(1) Gripping of the work piece by the feed jaws.
(2) Relieving of the vertical tie bars.
(3) Advancing of the work piece along the sawing line.
(4) Locking of the work piece.
(5) Advance of the saw slide.
(6) Return of the saw slide.

During the saw cutting, the feed jaws open up and return to their initial position. The cutting length is very constant, since the stops for regulating the cutting length are well protected in the feed cylinder.

A semi-automatic riveting machine Type NP is shown in Fig. 34. The machine is mainly used for counter sunk headed riveting. The rivet head is first clinched and then pressed on to the flat sheet metal. The clinched head is pressed from the top by a rivet stamp. A curved disc is driven by a chaindrive from the eccentric shaft, a bolt fixed to the disc leading to the lower machine. This governs the rivet centreing, so that after loading the machine the rivet is centred through a guide and during riveting the rivet guide becomes flush with the rivet anvil. 15 rivets can be finished per minute. Round headed and mushroom headed riveting can also be done on this machine.

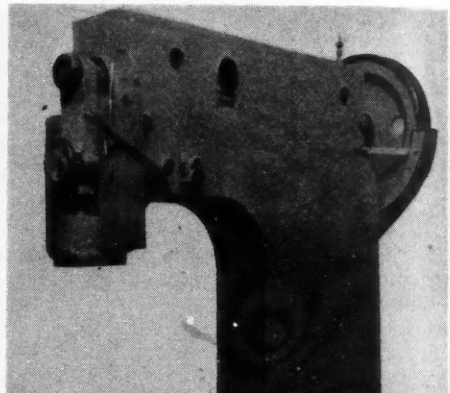


Fig. 34.

ENGINEERING FOR THE HIGHER ALTITUDE

By WELLWOOD E. BEALL. (From *Aero Digest*, Vol. 37, No. 4, October, 1940, pp. 52-57).

IN order to keep step with the rapid advancements in aircraft design, it is necessary that each airplane manufacturer develop much of its own specialised equipment. A major portion of the mechanical equipment for a modern airplane may be found either on shelves or in the laboratories of equipments. However, the introduction of cabin supercharging, the demand for power-boost for control surfaces, and the need for satisfactory temperature control equipment for aircraft cabins, presented problems which, because of their intimate relation to the airplane itself, made it necessary for the aircraft manufacturer to conduct independent research and development work.

The Boeing Stratoliner is the first example of supercharged cabin aircraft to be placed in service on commercial airlines. Actual service operations have proven the practicability of this new type of aircraft. " Overweather " operation without the physiological disadvantages of reduced atmospheric pressure and lack of oxygen is the first principle of supercharged cabin. A secondary advantage is to eliminate pressure variations within the cabin during rapid climb or descent of the aircraft.

Oxygen requirements at high altitudes are well defined by medical science. It is well established that active crew members on either military or civil aircraft should be supplied with oxygen when operating above 12,000 ft. altitude for extended periods, and at all times when flying above 15,000 ft. The use of modern oxygen equipment has proven very successful. However, due to the weight of oxygen storage equipment, it has always been used sparingly and, therefore, limited to high altitude flights of relatively short duration.

The ultimate solution of this problem is the supercharged cabin, wherein comfortable altitude pressure may be maintained when operating above 20-30,000 ft. Then, if by suitable means the altitude pressure within the cabin is maintained at a constant absolute value, the objectionable pressure changes experienced in unsupercharged aircraft will be eliminated. By reference to any book on anatomy, it will be seen that the middle ear is an air-filled cavity, vented through the nasopharyn which is just above and behind the soft palate. During the rapid climb, with the resulting decrease in atmospheric pressure, the pressure in the middle ear is normally relieved through the Eustachian tube at regular intervals. Tests have shown that this cycle occurs at approximately 425 ft. intervals for all ordinary rates of climbs. However, during an extreme rapid decrease of pressure, the Eustachian tube remains open throughout the initial decompression period. Any mention of rapid decompression brings up the question of " bends " caused by the release of nitrogen from solution in the blood stream. It is known that a sudden change to half of the pressure to which the body has become accustomed is not dangerous, while " bends " may be produced by a change to a pressure that is less than half of the original pressure. If we assume this to be true for the normal barometric range, the present day supercharged airplane is well within the range of safety. For example, if the pressure in the cabin of the Stratoliner were suddenly released when flying at 20,000 ft., the reduction would be from 9.25 to 6.7 lbs./sq. in., i.e. 27%. In experiments conducted by the Army on the ground, sudden pressure drops of 5 lbs./sq. in. were made without ill effect or even discomfort.

However, an increasing pressure, such as experienced during descent, may produce serious physical reactions which must be considered with operation of any aircraft. The process of ventilating the middle ear when the atmospheric pressure is increasing is more involved than during a decrease of pressure. For extremely low rates of change, pressure equalisation is more or less involuntary, since the Eustachian tube will permit a gradual inflow of air at low difference pressures. At greater rates of pressure changes the Eustachian tube will remain closed, and tends to collapse farther with any increase in difference of pressure between atmosphere and the middle ear cavity. This pressure difference can usually be relieved by voluntary action of the Eustachian muscles, except when the tube is blocked by an infection, such as a cold or in the event that the pressure difference is allowed to become excessive before the tube is voluntarily opened. The actual pressure difference at which muscular action fails to open the Eustachian tube has been found by test to be between 1.5-1.75 lbs./sq. in. Airplane operators have established a practice of limiting descents to less than 300 or 400 ft./min. to allow inexperienced passengers to ventilate properly the middle ear. Thus, it would seem that the problem is more one of annoyance rather than of any real danger. However, if the pressure difference is allowed to exceed the critical limit, the effect is sometimes painful and lasts several hours.

Pressure cabin aircraft with control equipment provide the most satisfactory solution to this problem. The foregoing analysis of physiological problems indicates that the supercharging equipment should be so designed as to maintain a normal atmospheric pressure within the cabin throughout the operating range of the aircraft. However, this condition is limited by other factors. The maximum allowable difference of pressure between the sealed compartment and the external atmosphere is determined by structural limitations. Weight and cost are the primary factors which govern the design of the structure. Ventilation requirements will further influence the design conditions, since the power available to drive the cabin supercharger is usually fixed. The question of ventilation rate may be varied somewhat to suit other requirements. Army tests have shown that the minimum rate per person necessary for normal respiration is approximately 5 cu. ft. of air per minute. Ventilation

practice indicates a much higher rate to be desirable for removal of the normal body odours. For aircraft work, a selection is usually made between 10-25 cu. ft./min./person. In the design of the Stratoliner, thorough considerations of all governing factors resulted in what has proven to be a highly satisfactory aircraft supercharging system.

An analysis of upper level conditions indicated that an operating altitude of 20,000 ft. would be sufficient to permit "overweather" operation on approximately 90% of the proposed schedules. Thus, the final decision on supercharging requirements was then based on the fact that oxygen is not necessary during operating up to 12,000 ft. altitude pressure. Cabin pressure events were arranged in such a manner as to maintain a cabin altitude pressure of 12,000 ft., when the aircraft is flying at 20,000 ft. level. This condition is maintained with a maximum cabin differential pressure of 2½ lbs./sq. in., which is not severe from a structural standpoint. The final specifications were drawn up on the basis that supercharging should start at 8,000 ft., and the altitude pressure at this level maintained in the cabin up to an actual altitude of approximately 15,000 ft., where the 2½ lbs./sq. in maximum pressure difference was reached. Then above 15,000 ft. the maximum pressure difference was to be maintained up to 19,000 ft. or over.

The experimental control unit consisted of an inlet valve which regulated the flow of ventilating air from the blower to the cabin at a predetermined rate, and an outlet valve to control the discharge of air from the sealed compartment and thereby maintain cabin pressure at the desired value. The two valves, although independent in action, were combined in a single compact unit to conserve weight and simplify the installation.

Regulation of the inlet air is by a rate of flow sensitive device, consisting of a throttling valve, operated by an air motor, flow piston and a metering pin which regulated the forces acting on the air motor, and a venturi connected to the flow piston in such a manner that the suction forces acting on the piston in opposition to a spring force operated the metering pin. The operation of this valve is such that any variation in flow through the venturi causes a corresponding change in the suction forces opposing the spring load on the flow piston. The resulting motion of the flow piston and the metering pin then opens or closes a port connecting the upper side of air motor piston to blower pressure.

The outlet valve is designed in such a manner as to be sensitive to an absolute pressure equal to an altitude of 8,000 ft. and to a difference of pressure between the cabin and the outside atmosphere of 2½ lbs./ sq. in. The unit consists of a throttling valve and air motor, an absolute pressure sensitive element and a differential pressure control. The absolute pressure control consists of an evacuated syphon bellows which expand with decreasing pressure and actuates a metering valve, thereby regulating the forces acting on the air motor piston and throttling valve assembly. This control closes the outlet valve and increases the cabin differential pressure at a rate of approximately 318 lbs./ sq. in./1,000 ft. for any altitude above 8,000 ft. Then at 15,000 ft. the maximum allowable difference of pressure of 2½ lbs./sq. in. is reached. The differential pressure control consists of a piston vented on the underside to cabin pressure and on the upper side to atmospheric pressure, a spring calibrated to restrain motion of the piston until the maximum differential pressure is reached, and a metering pin, which is operated by the piston and regulates the forces acting on the outlet valve air motor.

"Strato-Chamber" Devised.

Subsequent tests were conducted, both during actual flight, and by means of a large "Strato-Chamber" especially designed for testing cabin pressure control equipment. The complete altitude chamber consists of a 3 ton steel tank, 12 ft. long and 5½ ft. in dia., fitted with pressure-tight doors at each end, and divided into two compartments which are interconnected by a third airtight hatch. One of these represents the cabin of the aircraft, and contains all the instruments, instrument board and controls. The second compartment represents the atmosphere surrounding the sealed cabin and contains much of the altitude conditioning equipment, such as refrigeration to control the temperature of air, steam jets and chemical equipment to produce the effect of clouds and extremely dry air, and an auxiliary blower to stimulate the effect of ram pressure in the supercharger inlet duct. During

actual test, it has been possible to maintain standard atmospheric temperature changes in the chamber during a stimulated climb from sea level to 25,000 ft. at a rate of climb of 1,000 ft./min.

Strato-Chamber Altitude Control.

The altitude control for the Strato-Chamber consists of a heavy duty vacuum pump, driven by a 7½ h.p. motor, a small supply tank and control valves. The vacuum pump is operated continously during altitude test and maintains an absolute pressure of from ½ to 2 lbs./sq. in. in the supply tank. By means of valves, one outside and another inside of the cabin chamber, air is bled from the atmospheric chamber to the supply tank at a rate necessary to produce any desired rate of climb or level flight condition. A flow measuring nozzle is included in the system to indicate any leakage through the doors or joints.

The inlet to the supercharger is connected to the atmospheric chamber by an inlet duct. If the effect of ram, due to forward motion of the airplane, is desired, an auxiliary blower is installed in the chamber to produce the required pressure at the inlet. The supercharger discharge is connected to the cabin chamber by another duct, which also contains the intercoolers and heat exchangers. A flow measuring nozzle is installed in this duct to record the rate of flow through the system.

Duplicate instrument boards, one outside and the other in the cabin chamber, are equipped with a full complement of instruments. Provision is made to photograph either board, as is done during flight test of the airplane. In order to stimulate normal cabin leakage, the two chambers are interconnected by duct and valve systems, operable inside and outside, and the flow through these systems is regulated to a value equal to that expected during normal operation of the aircraft.

Two Ford V-8 power plants are provided to drive the cabin chamber supercharger. A gear box is provided with each power plant so that the engine speed will be equal to the aircraft engine r.p.m. during actual flight.

At present the Strato-Chamber is being operated on an average of 8 hours a day and 5 days a week, to keep abreast of the work on hand. Since all of the Stratoliners have been delivered, and only a few spare control units remain to be adjusted, the majority of the work being done is in the nature of substratosphere research and development of new equipment.

ESTIMATING THE EFFICIENCY OF LIGHT A.A. GUNS, WITH RESPECT TO THEIR SIGHTING DEVICES

By CAPTAIN H. BAASCH. (From *Flugwehr und Technik*, Zürich, Vol. 2, No. 8, August, 1940, pp. 176-177).

THE comparatively small number of aircraft shot down by anti-aircraft artillery during the last war aroused a disregard towards the efficiency of A.A. batteries. The following table shows the average number of shells required to destroy one aircraft in various countries during the last war :

Year	Germany	France	England
1915	11,885	—	—
1916	9,889	11,000	—
1917	7,418	7,500	8,000
1918	5,040	3,200	4,500 (1,500)

Reports published after the war show, however, that some of the A.A. batteries were well known by the pilots to be considerably more effective than the others, in consequence, the location of these batteries were avoided if possible. The tendency among the pilots became to disregard only those batteries which, for some reason, could not make full use of the possible effectiveness of their guns.

In view of this, the greatest care was taken to give the best training to the troops concerned, and also to improve as far as possible the instruments of the sighting and other supplementary devices. The manufacturers were given a free hand to carry out the necessary improvements, and as a result, greatly differing instruments were produced. The comparison of their respective efficiency is rather difficult, and many requirements, apart from the accuracy of the

sighting, i.e., cooling of the gun, quick adjustability, etc., must be taken into account.

A satisfactory way of comparing the efficiency of the implements which otherwise satisfy the requirements of the troops, is to express their efficiency as a percentage of the maximum possible attainable. In what follows, we wish to investigate the maximum possible probability of hitting, considering also some minor errors in the sighting process, and also how these results can be employed to compare their respective efficiency.

The formulas given below form the basis of our calculations and are well-known in text books on ballistic. Firing against a flat surface is assumed, and the probability of hitting can be calculated if the distance of the situation of the middle of the impact from the centre of the surface is known, hence:

$$(1)\quad W = \frac{h_1}{\sqrt{\pi}} \int_{\omega_\sigma - \xi}^{\omega_\sigma + \xi} e^{-h_1^2 x^2}\, dx \cdot \frac{h^2}{\sqrt{\pi}} \int_{\omega_\gamma - \eta}^{\omega_\gamma + \eta} e^{-h_2^2 y^2}\, dy$$

The aircraft can be replaced for convenience by a rectangle with dimensions 2m. by 6m. Hence, the values of ξ and η become 3 m. and 1m. respectively.

When estimating the efficiency of an anti-aircraft battery, we will not confine the calculations to individual shots, but we take into account the characteristics of the automatic gun as such and the method of firing as well. As a rule the light A.A. guns do not follow the aircraft continuously, but start firing with a considerable lead, and keep on firing with fixed sighting elements until the shells explode behind the aircraft. The plane receives a hit if the firing is fast enough and, of course, if the sighting elements were correct.

Mathematically, this case is equivalent to estimating the probability of firing on a strip, (Fig. 1). The width of the strip being in this case 2η. Hence:

$$(2)\quad W_\gamma = \frac{h}{\sqrt{\pi}} \int_{\omega_\gamma - \eta}^{\omega_\gamma + \eta} e^{-h^2 y^2}\, dy$$

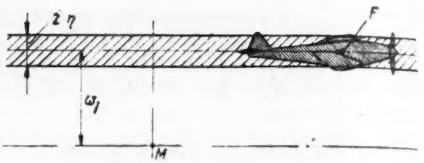


Fig. 1.

The value given by W_γ is halved in order to allow for the possiblity of the plane to slip through in between two consecutive shells, which might happen if the firing was not fast enough.

Equation (2) can be expressed in a more convenient form:

$$W_s = \tfrac{1}{4} \left[\Psi\left(\frac{\omega_\gamma + \eta}{w}\right) - \Psi\left(\frac{\omega_\gamma - \eta}{w}\right) \right]$$

where w is the half of the 5% dispersion of projectiles. The values of Ψ are given in tables. The probability of hitting corresponding to 2 —4‰ dispersion, curves for sighting errors up to 5‰ are given in Figs. 2 and 3. From these graphs the great effect of sighting errors on the efficiency is obvious.

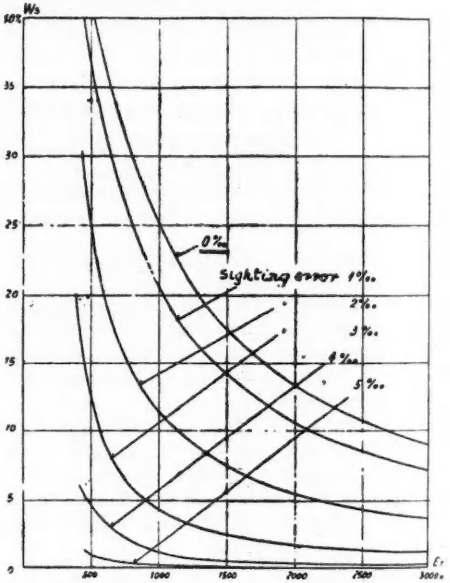


Fig. 2.

The group of curves given in Figs. 2 and 3 is most helpful in estimating the effectiveness of various sighting devices. As an example, let us consider the case, where the price of a particular sighting device, having an average sighting error of about $5^0/_{00}$, is 2,000 Swiss francs. Again, the price of an improved device, which has a sighting error of $4^0/_{00}$, is 10,000 Swiss francs. Assuming further a dispersion of projectiles of $4^0/_{00}$, then from the table 2 we can see that at a firing range of 1,000 m., the probability of hitting is 3.4% in the first case, and 5.5% in the second case. This means that under similar conditions, the saving in shells amounts to 38% with the improved device. Hence, if 400 shells are fired on an average day, the saving is 152 shells per day. If the price of a shell is 12 Swiss francs, then the cost of the improved device is repaid after only four and a half days use.

Not only the importance of effective sighting devices can be thus seen, but also the great importance of correct sighting and handling of the guns.

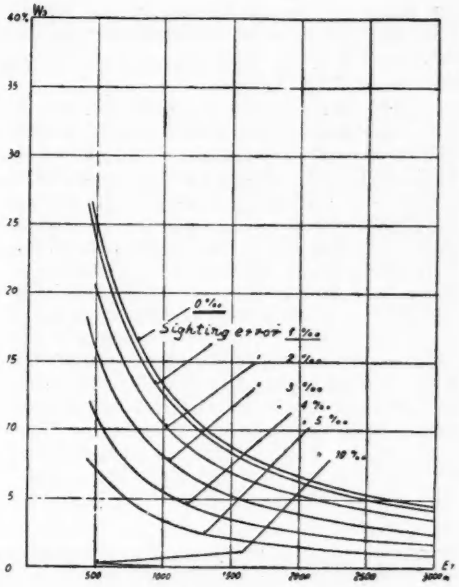


Fig. 3.

DROP TEST OF SHOCK ABSORBERS

By W. C. TRAUTMAN, Chief Designer, Bendix Aviation Ltd. (From *Aviation*, Vol. 39, No. 10, October, 1940, pp. 55 and 94).

A GOOD shock absorber does absorb the energy of "bounce" landings, and it contributes to the comfort of passengers, by smoothing out flying field irregularities while the plane is taxi-ing.

The fundamental mathematical precepts of shock absorber design are fairly well established. However, they involve the use of hydraulic coefficients which, in the final analysis, are derived from actual drop tests of the shock absorber and wheel combination. The drop testing of such combination is not quite as simple as it might appear, since it must be remembered that the total elapsed time of a drop test is less than one second. It is quite evident that, in order to record exactly what happened during this short period (1/40th to 1/60th of a second) the method of recording must not only be very accurate but also extremely fast. One of the first methods used for drop test recording was a motor driven drum, on which a pencil mechanically connected to the shock absorber traced the path of the shock absorber during the drop. This method suffered from lag due to the mechanical linkage involved and the inertia of the recording parts themselves.

Recently the engineering division of Bendix Aviation Ltd., has developed a high speed photographic recorder which has displayed a remarkable degree of accuracy. The equipment is based on the principle of ultra high speed camera, but the specific method used is novel and extremely simple.

Fundamentally, it functions as follows:

Light sources of "pin point" nature are provided in the fixed portion of the shock absorber, which is mounted in the drop test rig on the wheel axle and on two reference points on a fixed standard, which is usually placed about 12 in. to one side of the shock absorber. When the shock absorber and wheel combination is dropped in a darkened room

the lights on the shock absorber and wheel axle trace light paths, which have relative movement, one with the other, and with respect to the two fixed lights.

These light patterns are photographed as dots by using a motor driven shutter on the camera to break up the light source. To avoid congestion on the film caused by having these points fall in a straight vertical line, the camera is mounted on a rigid pivot and "fanned" in a horizontal plane during the time drop. This produces the exact pattern as shown in Fig. 1. A shutter speed of one picture every 1/240th of a second gives excellent results.

Each drop test is recorded on a single photo-negative. The completed negative graph will show four light-dot paths, one for the shock absorber, one for the wheel travel and a horizontal path for each reference lights. The stroke of the shock absorber or wheel can be determined at any time interval by comparison with the two reference lights. To facilitate the computation of the data the negative is placed in an accurate projector and the image of the graph is projected on a large piece of graph paper. The amount of magnification being adjusted to give an easily scaled distance between the reference points. These data are then extended until the curve showing the load factor developed at any point in the stroke, is found. A representative data sheet is shown in Fig. 2.

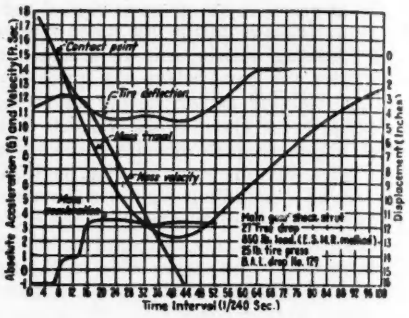


Fig. 2.

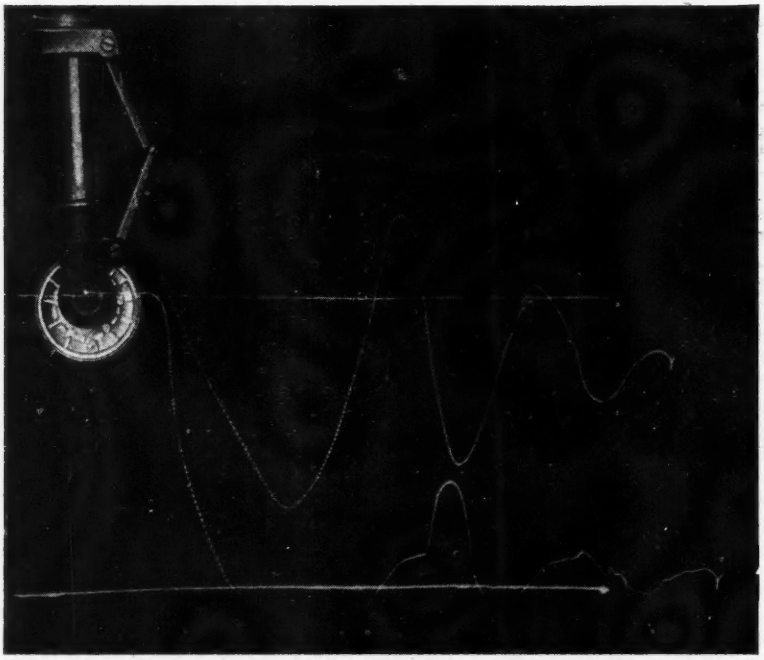


Fig. 1.

Very little similarity exists between the various drop test records owing to the tyre whose volume of air and initial air pressure differ; the softer the tyre the greater the rebound of the wheel. Thus, in spite of distinct advantages, low-pressure tyres have increased the problems of the shock absorber designer

"LETOV" DROP TEST MACHINE

(From *Letectvi*, Prague, Vol. 18, No. 11, p. 418 and No. 12, pp. 460-461).

Recording Energy Absorption of Shock Absorber and Tyre.

Fig. 1 shows the LETOV testing apparatus, designed for experimental work, which is also suitable for routine testing. The shock absorber wheel combination, together with the weight attached to it, weighs 4,000 kg. One day's work represents 400 drop tests, i.e. 40 tested shock absorbers.

It can be seen from the figure that the main construction is of standard design ; two parts, however, are worth mentioning.

First, the dynamometer as shown in Fig. 2. The landing gear is dropped on to the table,

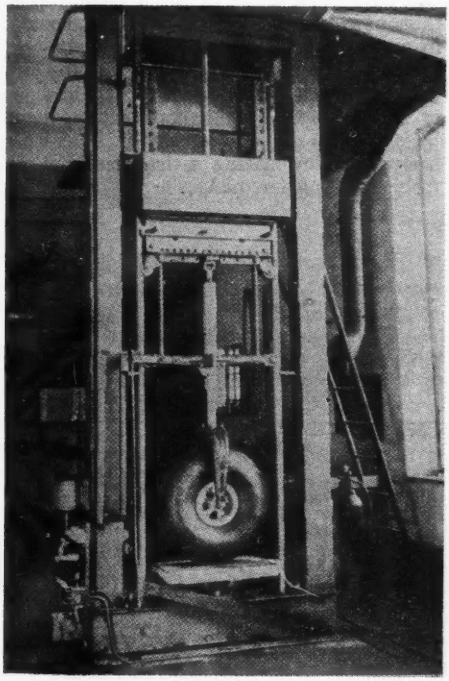
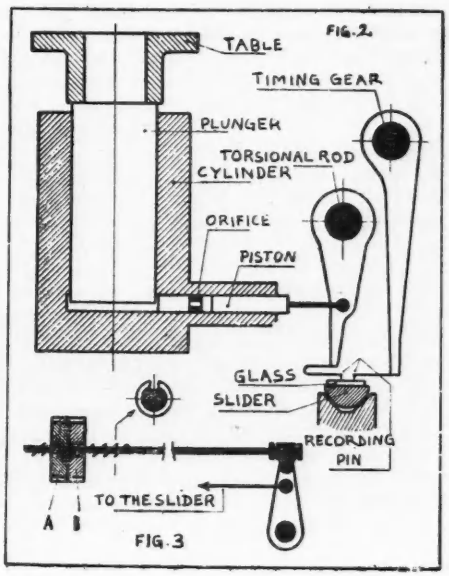


Fig. 1.

which is mounted on top of the plunger. The clearance volume is connected through a short channel to a small cylinder. The piston rod is connected to a lever, which, in turn, is rigidly attached to the end of a rod made of special steel, the torsional oscillation of which acts as a spring. This arrangement ensures a minimum inertia force, and hence, a maximum frequency (at about 100 osc. per sec.) of the whole system is obtained. The above mentioned lever governs the recording pin, which records the oscillations on a microscopic glass plate of dimensions 26 × 76 mm. The graph obtained on this glass gives very clear results when photographically magnified. In order to obtain smooth recording, damping is provided by means of a small orifice between the plunger and the piston. Another recording pin is connected to the timing shaft, which is driven by a synchronised electric motor, and registers the time intervals on the plate.

Fig. 3 shows in sketch a differential screw



mechanism, which was used with much success to register the relative longitudinal movement between the cylinder and piston of the shock absorber, plotted against the wheel travel. The principle of operation is as follows :

Pinions A and B each engage a rack. Pinion A is provided with a screw thread. The screwed shaft is connected to pinion B by means of a key, for which purpose a long keyway is cut in the shaft. The rack which engages pinion A is connected to the cylinder of the shock absorber, whilst the rack engaging pinion B transmits the movements of the wheel axle by rotating the screwed shaft. Thus, the differential movement between A and B is transmitted to the lever which operates the slider carrying the microscopic glass plate. On the resulting graph the deformation of the landing gear is plotted against the force and the time.

MARINE GAS TURBINES

(From BROWN BOVERI REVIEW, BADEN (Switzerland), January-March, 1940).

SWISS engineers, according to the article, have been investigating the possibilities of the gas turbine for driving ships. Although conditions for marine plants are different from those for land plants, the gas turbine holds interesting possibilities for naval and merchant-marine purposes.

In a warship, the main characteristics demanded are that the propelling plant be light in weight, take up little space, and, if possible, be simple to operate and maintain. Even in its present stage of development, the gas turbine has all these characteristics. In a plant of this type, as shown in Fig. 1, there would be no boiler, and no pumps, condensate pumps, and feed-water pumps. However, in order to achieve a low fuel consumption at cruising speed, it would be necessary to put in Diesel engines for cruising and to start the gas turbines.

Marine gas-turbine plants differ from land installations mainly in having to be reversible. One gas turbine drives the compressor. Since the compressor set delivers no mechanical power, its gas turbine only uses 75% of the compressed air from the compressor. The remaining 25% is led to the driving turbine. In going astern or in manœuvering, the compressed air would be taken from the compressor without its being heated, thereby making it unnecessary to use turbine blading designed for high temperatures.

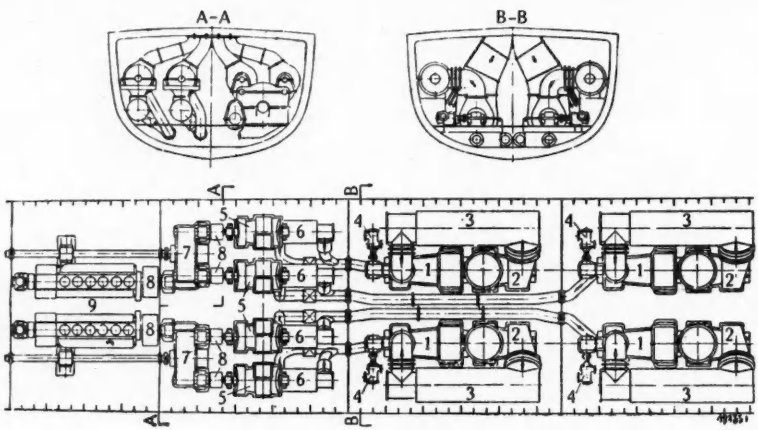


Fig. 1.

At full speed ahead, the air is brought up to full temperature in a combustion chamber, which forms part of the driving turbine, before being led to the ahead blading stages of the turbine. If propellers with adjustable blades are ever designed, the astern blading could be eliminated and it would not be necessary to separate the compressor set from the driving turbine, thereby simplifying the whole plant.

For merchant ships, which travel two thirds of their useful lives at about 70% of their full speed, only gas turbine plants with an air preheater should be considered, since the use of a preheater makes possible a thermal efficiency of 23%. This type of installation is cheaper to operate than a Diesel engine in those countries where fuel oil is cheaper than Diesel oil.

LUBRICATION AND ITS RELATION TO MAINTENANCE

By C. W. PHILIPS and R. A. BARTA. (From *Iron and Steel Engineer*, Vol. 17, No. 9, Sept., 1940).

A LUBRICATION engineer, or any planned lubrication programme, is a definite maintenance aid in any plant. A definite understanding on how lubrication problems, or sources of trouble pertaining to lubrication, are to be handled, can be made to show improvements when careful planning and follow-up is involved.

Various preliminary steps are involved in selection of a machine or mill which is being purchased to perform a certain amount of work. The future maintenance on any machine involves an intelligent selection of all steps concerned. The method of lubrication is an important step, bearing on the future maintenance costs.

An initial survey of lubrication on a machine should include :

(1) Checking points drilled for lubrication to see if lubricant will actually reach bearing surfaces.

(2) Oil tight cases and lubricant retainers.

(3) Accessibility of points to be hand-greased.

(4) Advisability of installing greasing or circulating oil systems.

(5) Checking design of these features for most economical use of lubricants.

(6) Clearance and replacement of bearings.

Selection of lubricants usually leads to a comparison of manufacturer's specifications with standard lubricants already being used elsewhere in the plant. Too many lubricants usually leads to confusion and unnecessary handling.

In any old or new mill a definite method for determining the schedule of lubrication, indicating all points of lubrication and the personal responsibility for these, should be charted in a clear detailed manner. Definite results have been obtained where this procedure was used and carefully followed through. When chart lubrication is used, men are more interested in the lubrication of the mill and take greater care than they did previously.

A proposed schedule of lubrication may reveal weaknesses which may be used as a basis for recommendations to improve and obtain more efficient lubrication. These may involve :

(1) New grease systems.

(2) Improved hand greasing equipment.

(3) Mishandling of equipment.

(4) Defects in circulating oil systems.

The portion of maintenance cost due to lubricants, bearings and labour for lubrication is only a small amount of the total cost. A monthly record of lubricants and bearing cost in each department can give accurate indication as to where help is needed in the maintenance programme.

Crankcase oil for diesel locomotives can be maintained in a clean condition suitable for long periods of service. Efficient filtration of these oils has resulted in noted improvements in locomotive performance and maintenance. A viscosity curve can easily be developed for determining amounts of fuel oil dilution in crankcase oils.

Lubricants are one of the most expensive supplies used in the maintenance of machinery. Carefully supervised application of lubricants is the most important item in keeping these costs down. Automatic grease and circulating oil systems help out in this respect. Methods of reclaiming contaminated oils keep down oil consumption. Mixing of oils with extreme pressure bases results in highly effective and low cost extreme pressure oils.

AXIAL VIBRATION IN CRANKSHAFTS

By P. DRAMINSKY and T. WARMING. (From *Bulletin Technique du Bureaux Veritas*, May, 1940, pp. 13-16).

IT has recently been stated that important axial vibration took place in the crankshafts of some marine diesel engines of a certain power. This vibration was sometimes very violent, and, on one occasion, it brought on the rupture of the shaft.

It has been thought that vibration of that kind could arise from the direct action of firing pressures leading to an increase of the length of the cranks as a consequence of the bending of the journals. However, that axial vibration thus produced is small and of minor importance.

The conditions necessary to give rise to axial vibration of a destructive character are the following :

(1) The frequency of the axial vibration must coincide with or be very nearly equal to the frequency of the torsional vibration.

(2) There must be one speed of important torsional vibration.

It follows that the chance of dangerous axial vibration is very small in actual practice, for No. 1 condition is a consequence of bad luck, and No. 2 condition is seldom met, as far as shafts carefully calculated for torsional vibration are concerned.

Methods of calculating the torsional vibration frequencies are well known. On the other hand, calculation relating to axial vibration has not been, so far, so carefully checked.

A formula, which, as far as marine diesel engines of standard manufacture are concerned, has been found satisfactory, is given hereinafter. It gives the frequency of the axial vibration of the first mode, viz., in which the mode is located on the thrust main bearing, and the greatest amplitude at the front end of the shaft. The amplitude curve is very nearly a sine curve.

Referring to Fig. 1 notations, the formula sets up as follows :

Let f_c be the natural frequency

$$f_c^2 = \frac{\pi^2}{4} \frac{EF}{ML} \qquad (1)$$

Thus, the critical speed is

$$n_c = \frac{60\,W_c}{2\pi}$$

In which formula :

E=coefficient of elasticity.

F=cross area of the journals.

M=pm=masses of the crankshaft from the front end to the thrust bearing.

p=number of cylinders.

L=equivalent total length $=pl=p\,(l_1+l_2+l_3-l_4)$.

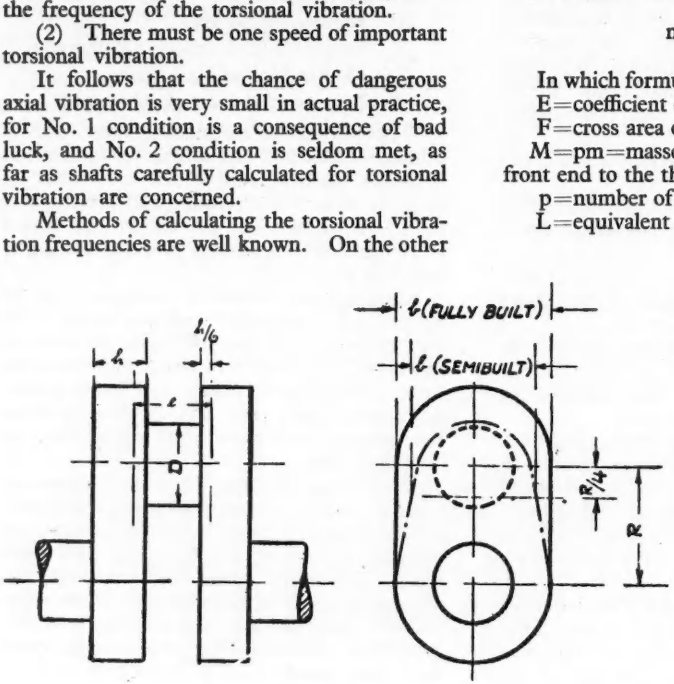


Fig. 1.

The length l_1 is that of an equivalent shaft which undergoes, when subjected to any given axial force, the same extension as that which would occur in one crank as a consequence of the bending moments created on the pin and webs by the action of the same axial force.

l_2, l_3 and l_4 are introduced in order to take into account : the extension directly due to axial stresses ; the extension due to shear stresses in

the webs; the fact that the shaft actually is a continuous beam.

The sum $1_2+1_3-1_4$ has been found to be practically negligible, so that the formula may be written thus:

$$1_1 = 8FR^2\left(\frac{8e}{\pi D^4}+\frac{R}{Ch^3}\right)$$

In the case of marine engines having fairly long crankshafts, the axial vibration frequency is usually about half-way between the frequencies of the first and second mode torsional vibrations. On the other hand, in engines having very short intermediate shafts, in which the first mode frequency is relatively high, this latter may coincide with that of the axial vibration.

For a better understanding of the way by which the torsional vibration may give rise to axial vibrations, let us consider a shaft having only two cranks, at an angle α to each other (Fig. 2).

The shaft is supposed to be fixed at the third bearing and to undergo a twisting moment directed as the arrow and acting upon the first journal. If journals 1 and 2 were allowed to move freely, the shaft would be twisted as shown in dotted lines. The new location of 1 and 2 journals would be as shown in Fig. 3. Length f represents the displacement of the journal, which results in each crank from the twisting of the crankpin and the bending of the webs.

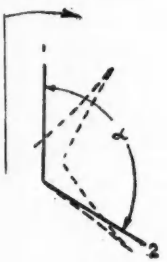


Fig. 2.

Fig. 3.

The applied twist does not give rise to any lengthwise alteration of the shaft. However, the journals being, in fact, not allowed to move crosswise and being compelled to stay in line, journal 2 must be moved transversely till it reaches the axis 1-3, viz., it must move over a distance:

$$f\cos\tfrac{1}{2}(180-\alpha) = f\sin\frac{\alpha}{2}$$

In every crank, the displacement may be reduced into components perpendicular to, and parallel with, the crankshaft. This latter component, whose value will be, in each crank,

$$f\sin\frac{\alpha}{2}\sin\tfrac{1}{2}(180-\alpha) = f\sin\frac{\alpha}{2}\cos\frac{\alpha}{2} = \tfrac{1}{2}f\sin\alpha$$

will lead to a marked alteration in the length of the shaft.

For both cranks together, this alteration will be the same as that of one single crank, fixed at one end, its other end being submitted to a force whose action would give rise to a crosswise displacement (in the plane of the crank) equal to $f\sin\alpha$ (see Fig. 4).

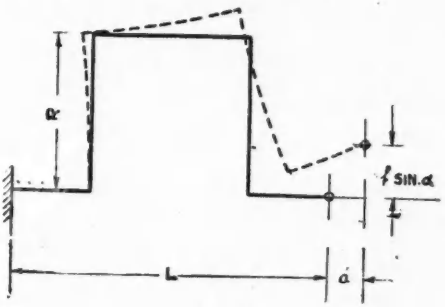


Fig. 4.

By calculation, the change of length may be taken as very closely equal to

$$a' = \frac{3}{4}\frac{R}{L}f\sin\alpha \qquad (2)$$

which reaches a maximum when $\alpha=90°$.

For crankshafts having more than two cranks, each group of two consecutive cranks will be successively dealt with, so that the total change in length will be:

$$a_t = \frac{3}{4}\frac{R}{L}\Sigma f\sin\alpha \qquad (3)$$

This formula contains one term for each group of two adjacent cranks, the number of terms being equal to the number of cylinders less one.

In each group, $\sin\alpha$ must be taken with its sign, that is reckoning always from the same

plane and in the same direction (it follows that α may be smaller or greater than 180°).

Equation (3) gives the total change of length of the shaft under the static action of the maxima moments of the torsional vibrations. In this latter case, the fluctuation in length may become very important.

According to the general theory of vibrations (damping being neglected), the amplitude at the front end of the shaft is :

$$a_1 = a_s F$$

as being the " static longitudinal amplitude," and F being the " dynamic magnifier "

$$\frac{n_c^2}{n_c^2 - n^2}$$

At a resonant speed, F depends on damping, the value of which appears to be very nearly the same as for torsional vibration.

Taking into account the various elastic reactions resulting from each term of (3), a_s may be calculated as follows :

According to equation (2), and c being the longitudinal rigidity of a given crank, the elastic reaction will be :

$$K = ca' = c\,\frac{3}{4}\,\frac{R}{L}\,f \sin\alpha$$

Let a_1 , a_2 , . . . be the actual longitudinal amplitudes of the isolated cranks.

Let a_1', a_2', ... be the differences between the amplitudes

$$a_1' = a_1 - a_2,\ a_2' = a_2 - a_3 \ldots .$$

The sum of the energies of the elastic reactions is $\Sigma \frac{1}{2} Ka'$ for a quarter of one cycle from the central position to the extreme one. (The amplitude curve has been taken as a sine curve).

The longitudinal amplitude a_s can be found by equating the potential energy to the kinetic energy, thus

$$\Sigma \tfrac{1}{2} Ka' = \Sigma \tfrac{1}{2} m\,(aw)^2$$

m being the mass of one crank.

Hence (a_1 is equal to a_s) :

$$a_s \Sigma K \Delta' = a_s^2\, w_c^2\, \Sigma m \Delta^2,$$

$\Delta_1 = 1$, $\Delta_2 = \frac{a_2}{a_1}$ being the relative amplitudes, and $\Delta' = \Delta_1 - \Delta_2$.

Whence :

$$a_s = \frac{\Sigma K \Delta'}{w^2 \Sigma m \Delta^2}$$

Replacing K by the expression given above, and noting that

$$w_c^2 = \frac{\pi^2 e}{4\,p^2 m} \text{ and } \Sigma \Delta^2 m = \tfrac{1}{2} pm,$$

one finds :

$$a_s = \frac{6p}{\pi^2}\,\frac{R}{L}\,\Sigma f \Delta' \sin\alpha$$

In actual practice, a_s will be calculated according to the values $f = \frac{1}{2} R\alpha'$ corresponding to the resonant point of the torsional vibrations (α' being the rotation of one end of one complete crank as compared with the other end).

As for the other speeds, a_s varies proportionally to the torsional amplitudes. At each speed, the value of a_s will be multiplied by the dynamic magnifier, which reaches its maximum at the resonant point of the axial vibration, whilst a_s reaches its maximum at the resonant point of the torsional vibration.

The a_1 speed curve is thus established. At the critical speed n_c, the question arises of the influence of damping. By taking the relative damping equal to that of the torsional vibration, the error is not likely to be an important one.

An example is then given, the figures arrived at by calculation being substantially the same as those measured.

The example shows that the calculation actually gives a maximum limit of the axial amplitudes. It clearly proves also that the axial vibration of a destructive character is actually a consequence of torsional vibration and not of the action of the firing pressures.

PORTABLE BRIGHTNESS METER FOR LUMINOUS PREPARATIONS

(From *Technical News Bulletin of the National Bureau of Standards*, No. 282, October, 1940, p. 85).

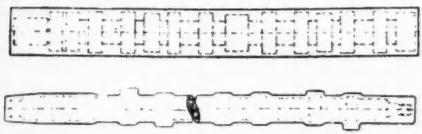
A COMPACT portable photometer for measuring the brightness of the luminous paints used on clocks, watches, and airplane instruments has been constructed from the design of Leon F. Curtiss, chief of the Bureau's Radioactivity Section. An arrangement of stainless steel mirrors is used to bring the light from the luminous paint into view adjacent to a luminous pattern from a reference lamp, the brightness of which can be controlled and calibrated. These mirrors do not tarnish and, therefore, require no protective coating which might cause undesirable reflections and also deteriorate with age. Since comparisons are made at fixed positions from the luminous paint and the reference lamp, the instrument is very compact and rigid. Therefore, it is not easily put out of adjustment by rough handling.

PRESENT METHODS OF TURNING CAMSHAFTS

(From *Die Werkzeugmaschine*, Vol. 44, No. 3, February, 1940, pp. 43-51).

WORKSHOP practice has shown that the quickest and most accurate way of mass producing engine parts is to employ special machines and automatics. Economical considerations are also in favour of these machines.

One of the common parts to almost every type of heat engines is the camshaft. The many stepped camshaft profile (Figs. 1 and 2) would require a long time, if turned on an ordinary lathe. The actual turning process consists of two operations : to turn the shaft, and to shape the cam-profiles.



Figs. 1 and 2

For the first of these operations various machines can be employed. Frequently, camshaft recessing lathes are used as shown in Fig. 3. The head stock is of very simple design, since it was found that very satisfactory results are obtained with a shaft speed of about 100 r.p.m. Turning could be finished very quickly if all points could be turned at the same time. However, if the camshaft is long compared with its diameter, then turning must be

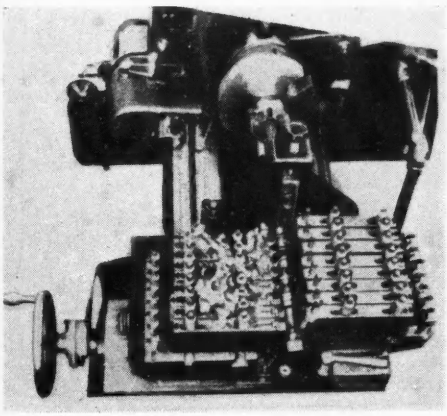


Fig. 3.

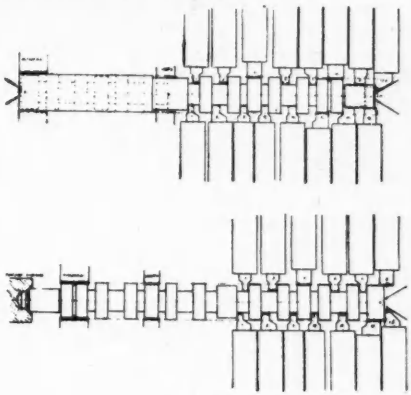


Fig. 4.

carried out in two or more operations as shown in Fig. 4, otherwise the camshaft would bend during turning. Steady plates would not give sufficient protection against this. But even if turning is taking place in two or more steps, it can be accomplished in about six minutes.

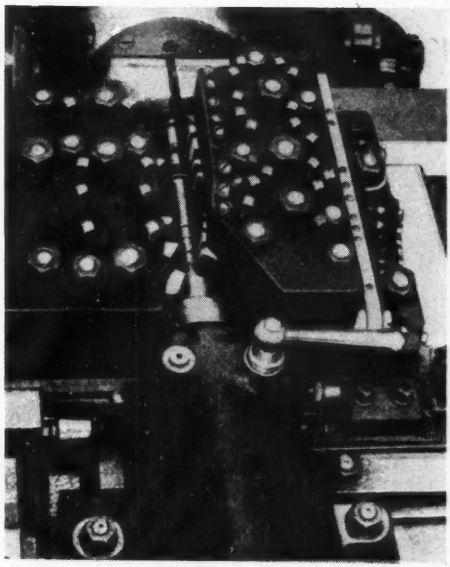


Fig. 5.

Alternatively, a multi-cut lathe (Fig. 5) can be employed. In this case, the camshaft is turned with a horizontal feed. This method is more useful when the distances between the adjacent cams are larger. The arrangement of the tools is shown in Fig. 6. The time taken with such machines is approximately 3 minutes.

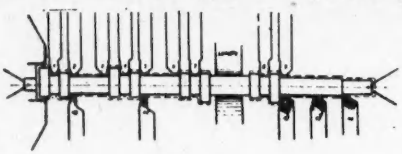


Fig. 6.

Next, the camshaft is worked on a " camprofile " lathe. Two types of these special machine tools will be described in the following. It must be mentioned already at this stage, that they considerably reduce the time of working and give a higher standard of accuracy, when compared with other types of machines. The two types are respectively: camprofile lathe with radially moved toolholder, and camprofile lathe with swinging toolholder.

The first type of these machines is illustrated in Fig. 7. Fig. 8 gives a schematic sketch of

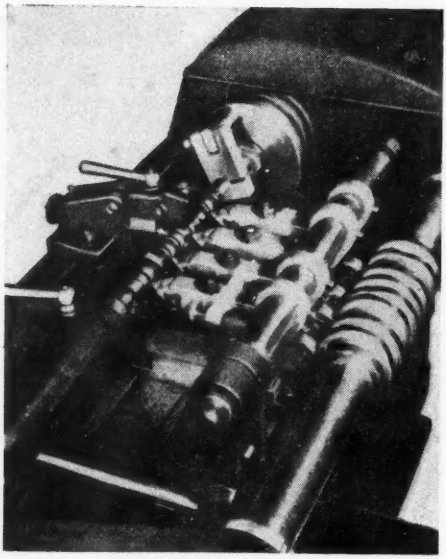


Fig. 7.

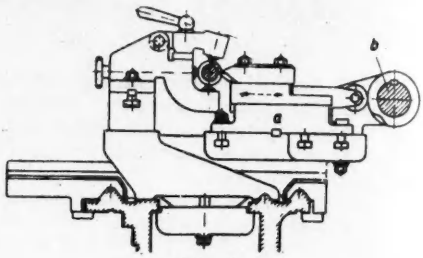


Fig. 8.

the same machine. The operation of the machine is easy to understand from these illustrations. The toolholders (a) are moved by the cams on the mastershaft (b). The hinged steady above the camshaft is for removing the work piece when finished or when inserted into the machine. The only disadvantage attached to this type of machine is that the cutting angle (Fig. 9) is continuously changing during turning. The relative positions of the tool and work piece is causing this change in the cutting angle, as shown in Fig. 10.

Quite different is, however, the operation of the swinging toolholder type of lathe (Figs. 11 and 12). The cutting angle remains constant during turning, since a second mastercam (c) is provided, which moves the toolholder in the vertical plane so that the tool is always at the right angle to the work piece (Fig. 13).

The turning process is the same with both types of machines. The tools are adjusted to the proper cutting depth. The feed must be very small (0.037-0.057 mm.) at a speed of 12-14 m./min. The slide of the toolholder is comparatively wide (Fig. 14) so that usually two adjacent cams cannot be turned at the same time.

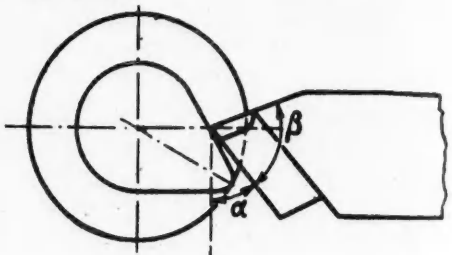


Fig. 9.

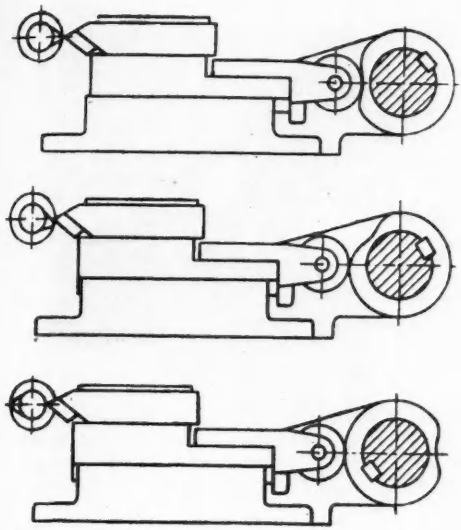


Fig. 10.

Since the camshafts are generally long, great care must be taken to keep the camshafts rigidly in position. For this purpose, steady

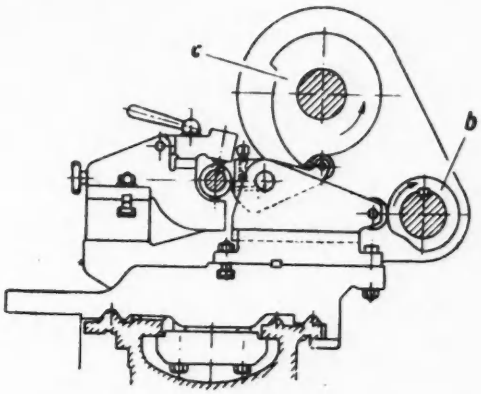


Fig. 11.

points are provided (Fig. 15), which are rigidly mounted on the main frame. Two spring-mounted rollers are arranged opposite to each other. By closing the upper steady, the rollers come into contact with the bearing places on the camshaft.

The toolholders are arranged at the back of the lathe and are fixed to the slides. The front part, being open, allows for easy access for

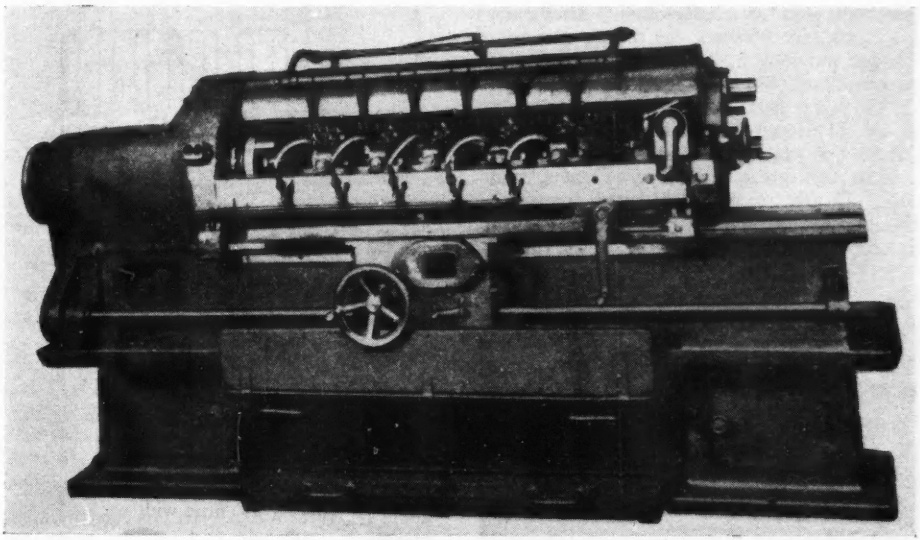


Fig. 12.

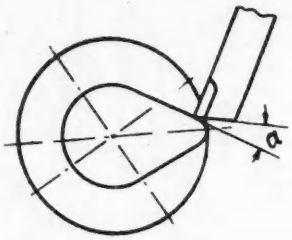


Fig. 13.

dimensional checking. A special adjusting control is used to bring the tools to the exact centre of the work piece.

The automatic feed can be adjusted according to the shape and size of the cams. The swinging toolholder type permits a deeper cut and gives a very clean finish.

Almost every type of camshaft requires a different working method, depending on its shape. The following are the principal working methods :

If the cams are wide apart, all cams can be turned at the same time (Fig.16). If the cams are close together, only every other cam can be turned (Fig. 17), and turning is finished in two operations. In this case, each toolholder is provided with two master-cams. After cams 1, 3, 5, etc. are finished, the master camshaft is moved and cams 2, 4, 6, etc. are turned with the second set of cams. If an additional cam (e.g. oil pump cam) has to be turned as well, the second master-disc corresponding to this toolholder is blank, i.e. circular.

With still greater number of cams, and hence, further reduced distances between adjacent cams the toolholder would push against the second master-cam before the cam is turned

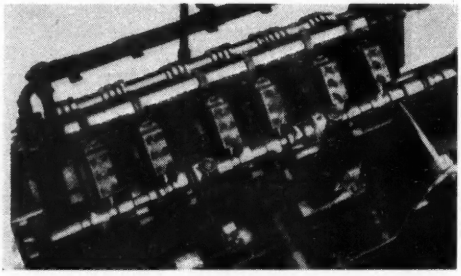


Fig. 14.

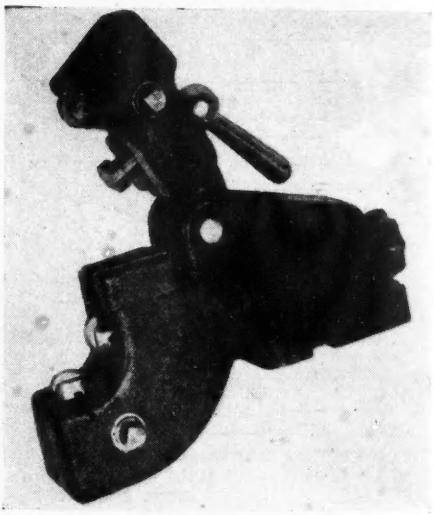


Fig. 15.

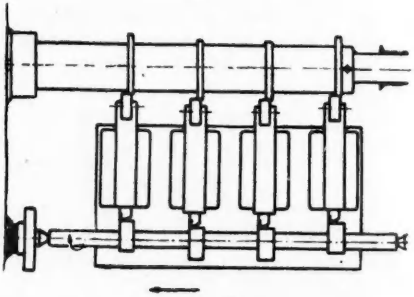


Fig. 16.

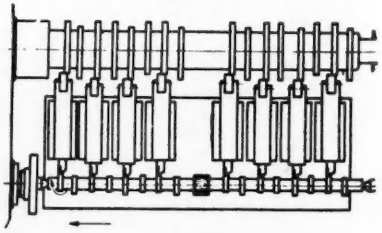


Fig. 17.

along its entire width. In this case the master camshaft must move with the toolholders (Fig. 18).

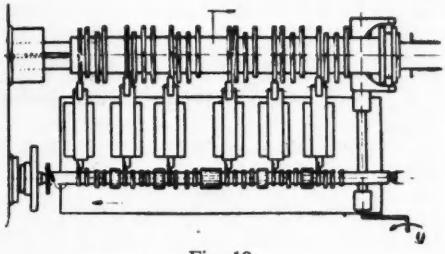


Fig. 18.

Turning is done in yet another way if only two toolholders are provided (Fig. 19). This is very economical if there are not too many cams to be turned. Consecutive sets of cams are machined in turn.

When, as occasionally happens, the order of the cams (inlet and outlet) changes from one set of cams to the adjacent set, three master cams must be provided for each toolholder.

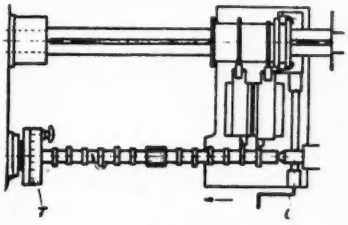
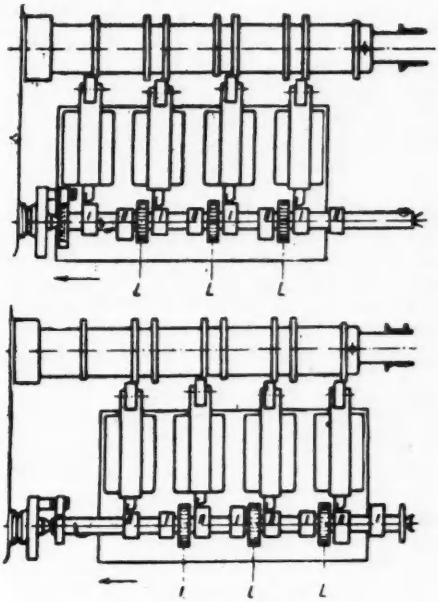


Fig. 19.

Naturally, the camshaft must be rotated by the appropriate angle for each set of cams.

Should the distance between two cams be too small (Fig. 20 and 21), then turning must be carried out in two settings. First, cams I are turned, then the work shaft is turned over and cams II are finished. There are also camshafts where the cams cannot be turned with a longitudinal feed. These are shafts where braking cams are situated next to the exhaust cams, and these are " blended " by a sloping part (Fig. 22). The trouble arises when the cam next to the eccentric has to be turned. The sloping shaft part does not allow turning from the left, and from the right there is not sufficient room to arrange for the necessary cutting depth. In this case, hand-controlled feed in the radial direction is used, i.e. the depth is taken off in several turns, and cutting takes place along the entire width of the cam.

Identical cams are usually turned with one toolholder, in turn. Special method must be



Figs. 20 and 21.

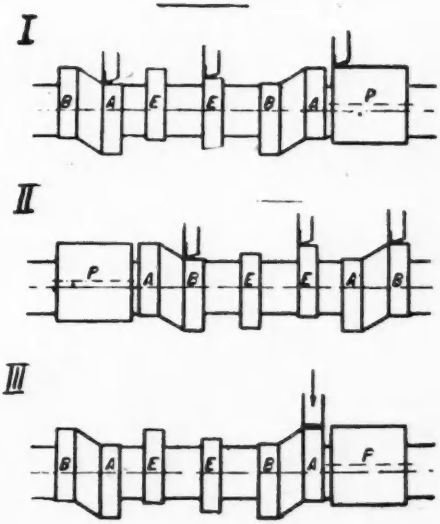


Fig. 22.

employed with very high cams (10 mm. and over). After roughly shaping the cams, the proper finish is obtained with a second turning operation.

TELEPHONE CABLE SUSPENDER

(From *Ericsson's Review*, No. 1, Vol. 1939, pp. 28-33).

RAPID development of aerial telephone network induced Ericsson to design a new type of cable suspender, which avoids various disadvantages of elder types used hitherto, especially chafing of the sheat of the cable. Though in recent years self supporting aerial cables have been introduced, the lead covered cable carried by means of a suspension wire is still in extended use. There are two types of suspenders: Suspenders which grip the cable firmly, but are free to move along the suspension wire, and suspenders which grip the suspension wire and let the cable move freely. Both types secure the maintenance of the original distance of the suspenders.

The new suspender of Ericsson belongs to the second type, which is chiefly employed in U.S.A. and in other countries outside Europe. Elder types show the disadvantage, that the movements of the cable caused by the wind and the variation of temperature give rise to wear of the lead sheat. The new type of suspender (Fig. 1) is made of bright, stainless spring band steel with the edges rolled down to prevent injury to the cable. Dimensions of the suspenders and of the cables for which they may be used, are given in the following table:

Suspender type	Width of the band	Diam. of cable	Type of suspension wire	Strength of suspension wire	Distance between suspenders
	mm	mm		tons	m
NF455/18	18	18	4.7 mm wire or 7×1.5 stranded	1.5	0.5
NF455/23	20	23	7×2.2 „	3	0.5-0.75
NF455/38	25	38	7×3.0 „	6	0.75

The ends of the suspender bands are rolled so as to form sleeves, which grip over the suspension wire. When being mounted the two sleeves are clipped on the suspension wire one over the other, so that pear shaped rings are formed, through which the cable is pulled in lengths of 500 to 600 yards. No tool is required for clipping the suspenders on the suspension wire (Fig. 1). The suspenders are

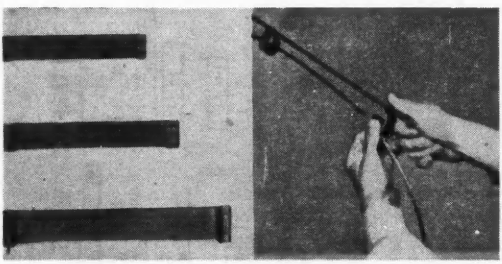


Fig. 1.

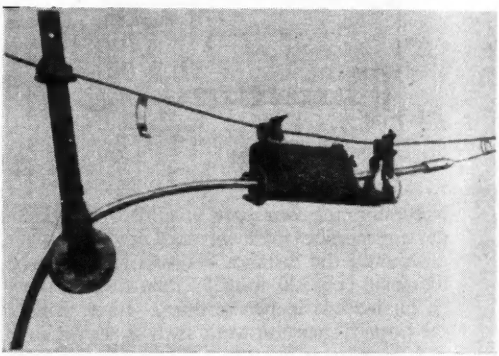


Fig. 2.

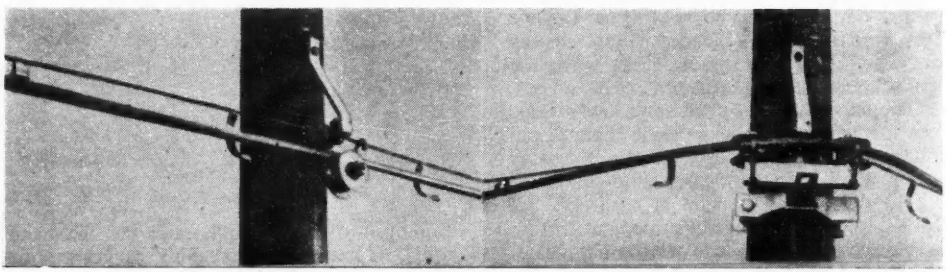


Fig. 3.

supplied flat, as this is most convenient for packing. Dimensions and form of the suspenders after clipping on the wire are such that wear of the cable sheat is prevented.

Before raising the suspension wire, spacing of the suspenders is marked by running a marking wheel along the wire temporarily run breast high. The circumference of the wheel is equal to the suspender spacing. The rim of the wheel is at one point provided with a hole, through which marking paint out of a container runs out every time the hole makes contact with the wire. The cable is pulled through the suspenders by using a pulling wire and a winch. It passes first a guiding in pulley (Fig. 2) and a lubricating box attached to the suspension wire so that a smooth pull is secured. Where necessary guiding rollers are temporarily attached at the poles (Fig. 3).

For stringing 550 yards of aerial cable a gang of 6 men is required for approximately 7 hours.

FAILURES IN OVERHEAD LINES WITH ALUMINIUM CONDUCTORS

By Dr. Ing. G. Lehmann, Dresden. (From *Elektrizitäts-Wirtschaft*, Vol. 39, No. 8, March, 1940, pp. 113-5).

After a thorough examination of the 110 kV overhead lines of the Saxonian State Electricity Works (ASW), which revealed that many wires, in some conductors even as much as 50%, were broken in the suspension clamps, it was decided in 1929 to equip all conductors of this 110 kV system with Hofmann damping reinforcements. Conductors with more than four broken wires in one clamp were replaced by inserting an appropriate piece of a new conductor. Since then, 1930/1, the regular examination of the lines showed no new failures, some sporadic breakings excepted. All these conductors of 150 mm² (250,000 circ. mils.) were strung according to a maximum tension of 8 kg/mm² at —5° C. with an ice loading of $0.18\sqrt{d}=0.716$ kg/m according to the rules of the VDE (that is 2,650 lbs. total tension or 11,378 lbs./sq.in. at 23° F. and an ice loading of cca. $\frac{3}{4}$" radial thickness). Only the first line with aluminium conductors erected in 1919 had been strung according to a maximum tension of 7 kg/mm² (2,320 lbs. total tension or 9,956 lbs./sq.in.) at the same loading conditions as indicated before. No broken wires had been found on this line up to to-day.

During the last ten years, there occurred in this 110kV system 6 complete breakings of conductors at tension clamps. As these failures caused most troublesome interruptions of the service, all tension clamps were examined, and consequently the point where the conductors are clamped was removed by one foot towards the free span by inserting lengthening parts in the tension clamps. In this way, the part of the conductor in which the full strength had been lost, due to breaking of wires, became eliminated.

For all new lines with aluminium conductors stringing according to a maximum tension of 7 kg/mm² (9,956 lbs./sq.in.) has been adopted

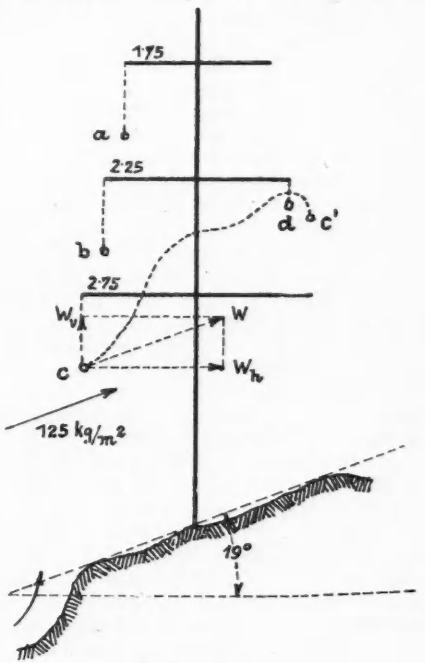


Fig. 1.

Aluminium conductor lifted by upward directed wind. a, b and c Aluminium conductors 95 mm², d stranded steel wire 50 m² ; C' position of conductor c when lifted by upward directed wind. Weight of conductor 0.257 kg./mm. ; upward directed force corresponding to wind pressure of 125 kg./m²; 0.257 kg./m.

on account of the good experience made with this lower tension. Besides, clamps were used, which suppress vibration. During the last two years, several lines have been strung with Preiswerk non-vibrating conductors. Up to today, of course, no sufficient experience with these conductors could be gained.

It follows from what is said above, that the danger of failures of wires in aluminium conductors has been greatly diminished during the last 6 to 9 years.

No wire failures have occurred in the 100 kV lines with copper conductors. In the medium high voltage system, some broken wires were found there and then in conductors of 50 mm² (98,000 circ. mils.), which were strung according to a maximum tension of 16 kg/mm² (22,757 lbs./sq. in.) at the loading conditions mentioned before. No steps were taken to deal with these failures, as no appreciable increase of the number of breakings could be ascertained. Many failures, however, occurred, due to vibration in the conductors of one 40 kV wood pole line, which in order to cut down the investments costs, was strung according to 19 kg/mm² (27,024 lbs./sq.in.). In spring, 1939, 48 broken wires on 46 poles were counted in the conductors of this line, half a year later, these figures increased to 119 failures on 63 poles.

During the last years several serious interruptions have occurred with overhead lines with aluminium conductors by the action of wind, which lifted the light aluminium conductors and caused direct contact between the wires. Such an incident on a 30 kV line is illustrated in Fig. 1. The aluminium conductors of 95 mm² (186,000 circ. mils.) on one side of the poles, which will carry later on a second circuit on the other side, are balanced meanwhile by a single stranded steel conductor. Such disturbances occur where, in consequence of the inclination of the soil, the wind is directed upwards. Sometimes the lifting action of the wind is increased by adhering ice or snow, which by a flat shape enlarges the surface the wind can work on. On places where such disturbances have occurred repeatedly, or where they may be expected, it has been suggested to attach to the conductors additional weights at a distance of 5 to 10 m from each other.

Fig. 2 shows the situation in a 100kV line, where upward directed wind has caused twice already a short circuit in the line, which was followed by the destruction of considerable lengths of the conductors. A change of the position of the towers or the attachment of additional weights to the conductors is under consideration in this case.

Another kind of failure, occurring in connection with thunderstorms, has been encountered in the clamps of lines with aluminium conductors. Conductors were found damaged inside of the clamps in the neighbourhood of towers, where probably flashovers had occurred. The conductors were molten within the clamp, and many wires were interrupted. It seems that these failures were caused by the oxidation of the aluminium surface of the wire which destroyed the good contact between the conductor and the clamp. When a flashover occurred on the insulator concerned, the high current, which passed from the conductor through the clamp to the ground, caused an excessive heating in the high resistance had to pass in the clamp, with the consequence of complete destruction of the conductor. It has been necessary to revise the clamps to restore good contact, where it had become insufficient, and to grease the conductor in order to delay oxidation. Where it was found necessary, the liners were renewed or replaced by a taping aluminium wire.

Similar difficulties were encountered with joints of aluminium conductors, in which the contact had become insufficient by oxidation. Such joints may become the cause of dangerous heating of the conductors diminishing their strength. Therefore, the Saxonian Electricity Works have eliminated the joints in some lines with very high load and have connected the conductors by welding.

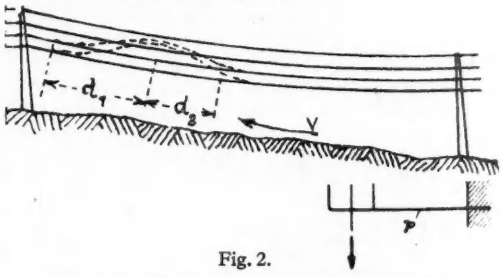


Fig. 2.

Failure on 100 kV-line caused by upward directed wind (velocity about 100 km./hr.). Four conductors were broken, from two of them a length of d_2= 20m. was missing, whereas all conductors showed marks of arcs on a length of approximately d_1=30 m. Both towers were heavily damaged.

OXYGEN BOOSTING OF AIRPLANE DIESEL ENGINES

By P. H. SCHWEITZER. (From *Mechanical Engineering*, Vol. 62, No. 10, October, 1940, pp. 719-722).

THE experiments carried out to investigate how Diesel engines will react to an increase in the oxygen-nitrogen ratio gave very important results. A $3\frac{1}{4}$ by $4\frac{1}{2}$ in. four stroke C.F.R. Diesel engine was used during the experiment, with a compression ratio of 15.8 to 1, and the engine speed was kept constant at 890 r.p.m. The schematic arrangement of the engine and the instruments is shown in Fig. 1.

The results of the experiment are shown graphically in Fig. 2. It can be seen that increasing the oxygen concentration from the normal 21% to 45%, the indicated horsepower increased from 4.3 to 6, the brake horsepower from 2.9 to 4.65, the maximum cylinder pressure from 700 to 780 lb./sq.in., and the specific fuel consumption from 0.43 to 0.59 lb. per I.H.P. hr. On the other hand, the maximum rate of pressure rise, which is an index of the shock loading of the engine, *decreased* from 88 to 60 lb./sq.in. per degree, together with the ignition lag which was shortened from $9\frac{1}{2}$ to 5 degree. The explanation for the favourable showing of the oxygen boost with regard to cylinder pressures and rates of pressure rise, is obviously the extremely beneficial effect the oxygen has on the ignition lag which was about cut in half by the oxygen addition.

With 55% oxygen concentration the power figures further increased somewhat, but above 50% oxygen, occasional detonation was experienced.

In Fig. 2 every point represents a great number of test readings. The comparison was made at optimum injection timing, which was determined for each oxygen concentration.

Using the absolute maximum load the engine is able to carry, is impracticable as under such load the engine soots up quickly. Instead of that, the comparison is based on

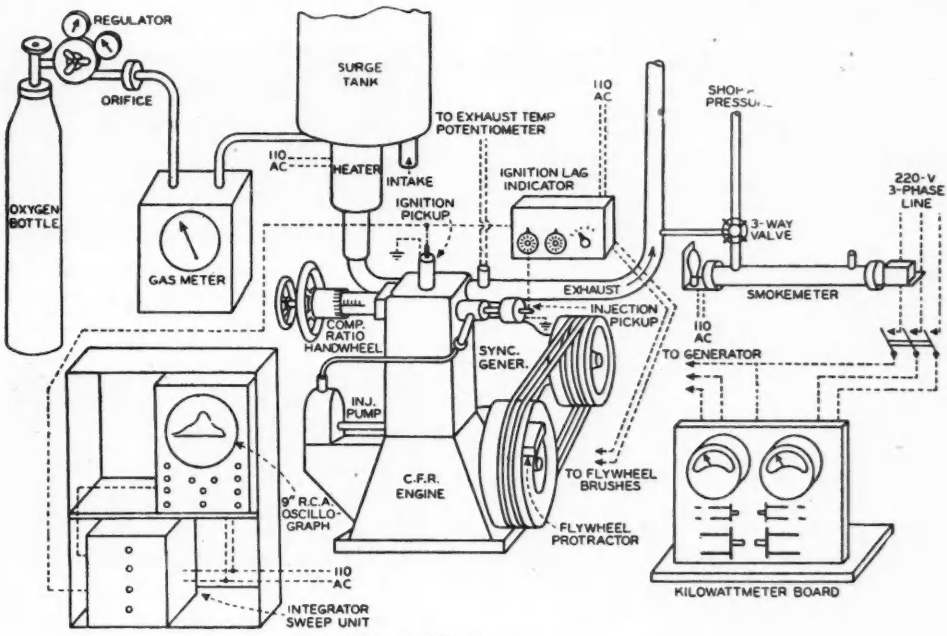


Fig. 1.

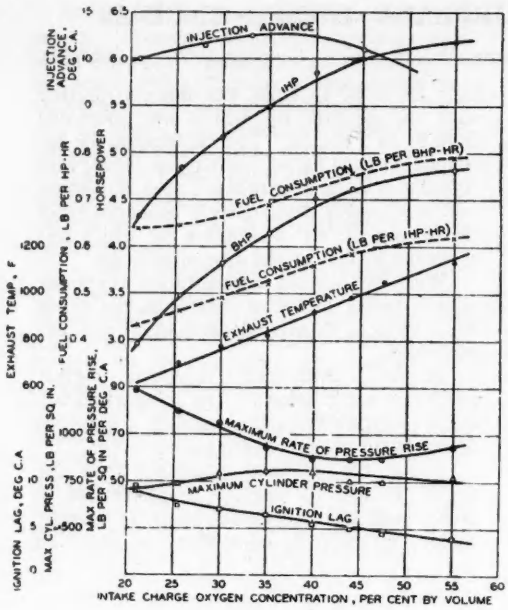

INJECTION ADVANCE
IHP
BHP
FUEL CONSUMPTION (LB PER BHP-HR)
FUEL CONSUMPTION (LB PER IHP-HR)
EXHAUST TEMPERATURE
MAXIMUM RATE OF PRESSURE RISE
MAXIMUM CYLINDER PRESSURE
IGNITION LAG
INJECTION ADVANCE, DEG C.A.
HORSEPOWER
FUEL CONSUMPTION, LB PER HP-HR
EXHAUST TEMP., F
MAX RATE OF PRESSURE RISE, LB PER SQ IN PER DEG C.A.
MAX CYL. PRESS., LB PER SQ IN.
IGNITION LAG, DEG C.A.
INTAKE CHARGE OXYGEN CONCENTRATION, PER CENT BY VOLUME

Fig. 2.
Test results, optimum timing conditions, 50% smoke.

"50% smoke." The load is increased until the Penn State smoke-meter registers 50% smoke density. This, incidentally, is the approximate limit of "clear exhaust." By simultaneously measuring power output and smoke, Fig. 3 was obtained, which permitted one to pick the points corresponding to 50% smoke.

The measured power outputs at optimum injection timing and 50% smoke would still be without practical significance, if they were accompanied by combustion pressures exceeding the limits of safety of rates or pressure rise causing intolerable knock. An R.C.A. 9 in. cathode-ray oscillograph supplied the information. Figs. 4 and 5 show typical oscillograms under normal and boosted operation. It will be noticed that the oxygen boost does not make the indicator card higher or steeper, only fatter.

For obtaining rate-of-pressure-rise figures directly, electro-magnetic pickups, instead of piezoelectric, were used. Such a card is shown in Fig. 6. Fig. 7 shows the same electrically integrated.

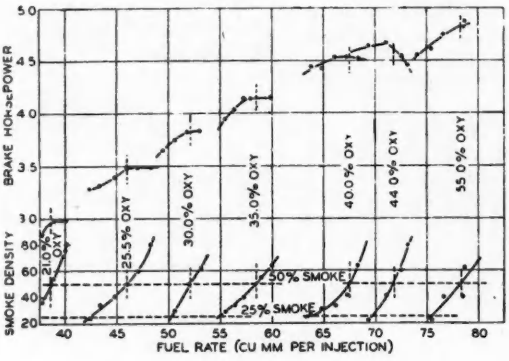


Fig. 3.
Brake horse power and smoke density at optimum timing.

Oxygen Helps Starting.

A series of reference fuels, from 29 to 72 cetane number, were tested with various degrees of oxygen boost to determine whether oxygen

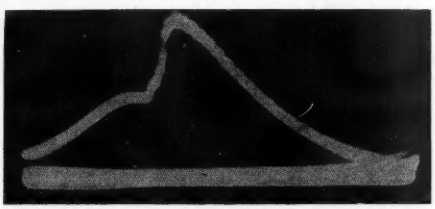


Fig. 4. Pressure diagram, Non-boost.
(2.7 bhp., 21% oxygen content, 32 cu. mm. of fuel per injection).

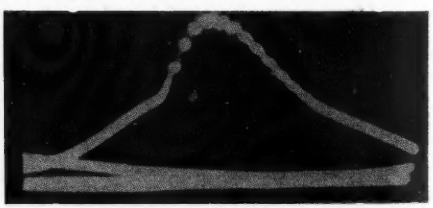


Fig. 5. Pressure diagram, Boost.
(4.65 bhp., 45% oxygen content, 71 cu. mm. of fuel per injection).

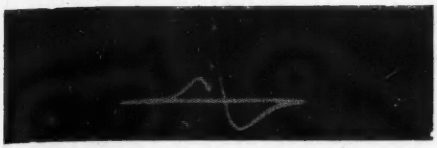


Fig. 6. Rate-of-pressure diagram.
(2.7 bhp., 21% oxygen content, 32 cu. mm. of fuel per injection).

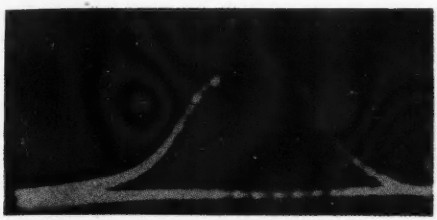


Fig. 7. Pressure diagram with degree scale.
(2.7 bhp., 21% oxygen content, 32 cu. mm. of fuel per injection. Bright dots are 10 deg. apart).

is a good starting aid. Their critical compression ratios were determined, which caused them to ignite within a period of six injections. The final results are shown in Fig. 8. It can be seen that the oxygen boost is equivalent to raising the ignition quality by 10 cetane number, when increasing the oxygen content to 45%. During operation of the engine, the same result was obtained, by raising the oxygen content only to 26%.

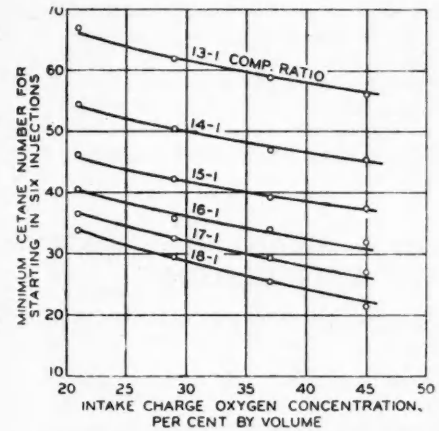


Fig. 8. Effect of oxygen on the minimum cetane number required for starting—C.C.R. determination (600 r.p.m.; intake air, 150F.; jacket-water temperature, boiling; starting points for six injections per trial).

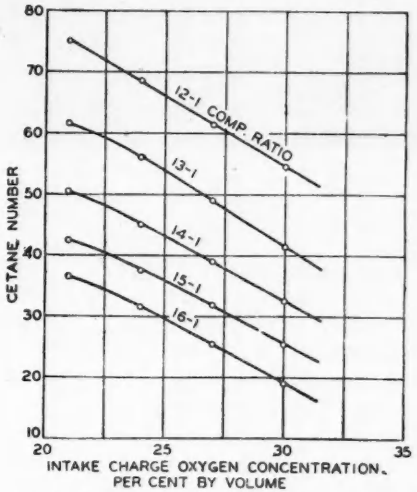


Fig. 9. Effect of oxygen on apparent cetane number, R.C.R. determination (15 deg. ignition lag; ignition, top dead centre; 900 r.p.m.)

Smoke Reduction.

Fig. 10 shows the effect of a small amount of oxygen on the smoke. 5% increase in the oxygen concentration is sufficient to make a black exhaust invisible.

Detonation.

At oxygen concentration above 50%, oc-

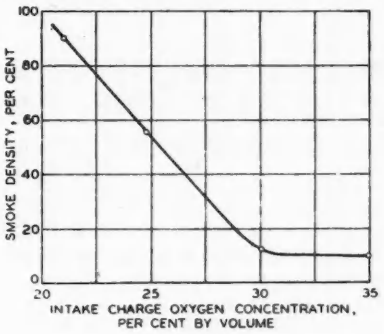


Fig. 10.

casional detonation was heard, but it lasted only for one or two revolutions at a time. Fig. 11 is the oscillogram of a detonating combustion superimposed on a number of non-detonating combustions. Below 45% oxygen concentration no detonation was observed.

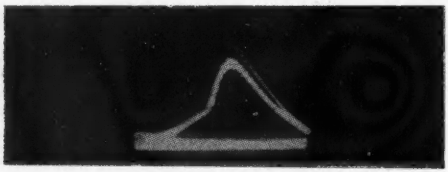


Fig. 11.

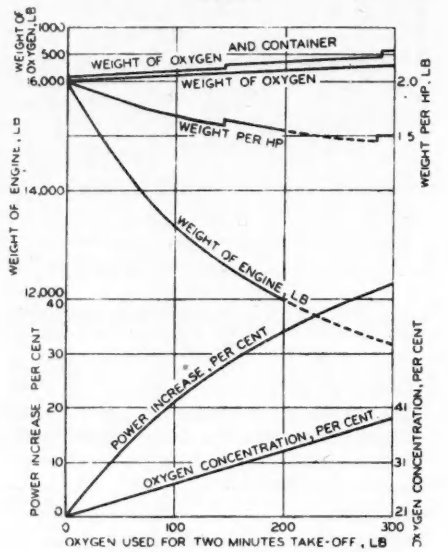

WEIGHT OF OXYGEN, LB
WEIGHT OF OXYGEN AND CONTAINER
WEIGHT OF OXYGEN
WEIGHT PER HP
WEIGHT OF ENGINE, LB
WEIGHT PER HP, LB
POWER INCREASE, PER CENT
OXYGEN CONCENTRATION, PER CENT
OXYGEN USED FOR TWO MINUTES TAKE-OFF, LB

Fig. 12.

Practical Conclusions.

Airplanes require about one third more power during the take-off than in flight. In Diesel-engined airplanes, the size of the engine could, therefore, be reduced 25% in feeding oxygen into the intake air during the take-off. Applying the results of the experiments to a transport plane, Fig. 12 shows the possible weight saving with various oxygen boosts. The curves are based on 6,000 cruising hp. and an estimated engine weight of 2 lb./hp.

The weight of the oxygen and container is 350 lb., while the possible saving in engine weight is 4,000 lb. The weight per take-off horsepower is thereby reduced from 2 to 1.54 lb. The calculation is given in the following table:

Normal horsepower	6,000
Take-off horsepower	8,000
Normal air consumption, lb. per min.	900
Normal oxygen consumption, 21% oxygen concentr. lb. per min. ..	189
Boosted oxygen consumption, 32% oxygen concentr. lb. per min. ..	289
Oxygen to be supplied lb. per min...	100
Weight of 8,000 hp. engine lb. ..	16,000
Weight of boosted 6,000 hp. engine lb.	12,000
Weight of oxygen for 2 min. boost lb.	200
Weight of container for 29 lb. of liquid oxygen lb.	150
Net weight saving by oxygen boost lb.	3,650
Weight per horsepower, non-boosted engine lb.	2
Weight per horsepower, boosted engine lb.	1.54

AIR-BLAST CIRCUIT BREAKERS AS A MEANS OF DEALING WITH SHORT-CIRCUITS BY AUTO-CLOSING

By G. BROCKHAUS. (From *Elektrotechnische Zeitschrift* [*ETZ*], February 29th, 1940, Vol. 61, No. 9, pp. 191-3).

IN recent years investigations were carried out in networks with earthed neutral with a view to "eliminate" short-circuits by means of short interruptions of the energy supply. The experience gained with this kind of short-circuit elimination suggested its application to insulated neutral systems equipped with Petersen arc suppression coils in respect to phase to phase faults. In certain networks thus equipped the number of breakdowns has been reduced by some 90%.

Reclosing of a breaker must be delayed until the ionized conducting gas is removed from the point of fault. The shorter the

duration of the arc, the less conducting gas is produced. Thus, by shortening the relay setting and the operating time of the breaker, the application of a shorter reclosing time is made possible indirectly.

The air-blast circuit breaker, which has been developed by the A.E.G.*, possesses some peculiarities, which render this type of breaker particularly qualified for this kind of work, as with these breakers the time from the moment the trip coil is energized until the arc is extinguished, is very short. So the duration of the arc at the point of short-circuit can be reduced to a minimum, especially if at the same time quick-acting relays are used.

In the air-blast breaker, the space between the contacts is cleared and cooled very effectively by compressed air. Consequently, there is no reason from the breaker point of view to delay reclosing, and this type is suitable for several breaks and makes, following upon one another almost immediately. The breaker may also be used then for the final break, when necessary. This is of great importance, as generally one will not be inclined and, of course, in many cases, not able to instal one breaker with the only purpose to deal with short-circuits by short interruption of the circuit and a second breaker for the final break, should the short-circuit not be eliminated by the short interruptions.

It is easy to adapt the air-blast breaker for both these functions. There is no difficulty to secure the compressed air necessary for repeated operations of the breaker. With some types of breakers, short-circuits may be eliminated by performing only a part of the regular operation of the breaker; e.g., with the so-called open-jet type air-blast breakers for voltages from 30 to 220 kV., which are designed on lines of isolator-switches (Fig. 1), breaking of the current is performed between the nozzles (1) by compressed air, which quenches the arc. At the same time, by a rotating movement of the insulators (2), the nozzles are separated from each other and brought in a distance appropriate to the voltage

When the breaker is used to eliminate a short-circuit by a short interruption, the

*See Stritzl, P. F., Oil-less Circuit Breakers, *Power Engineer*, January, 1934, and Stritzl, P. F., Oil-less Circuit Breakers, *Power and Works Engineer*, September, 1938.

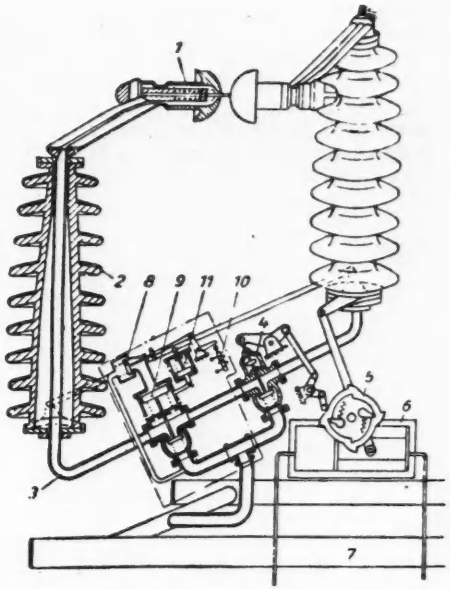


Fig. 1. Open-jet air-blast circuit breaker with equipment for "short circuit eliminating."

1. Nozzles
2. Insulators
3. Compressed air pipe
4. Valve for closing and opening of the breaker
5. Operating mechanism for valve.
6. Gear of the breaker
7. Compressed air tank
8. Auxiliary valve
9. Piston for actuating valve 8.
10. Magnet
11. Releasing piston

rotation of the insulators and the separation of the nozzles is not effectuated. The operation is restricted to the withdrawal of the contact rods into the interior of the insulating nozzles and the expelling and quenching of the arc by compressed air, so that it extinguishes at the next instant of zero current. For this kind of operation of the breaker, there is provided a special valve (8) for the compressed air, whereas valve (4) governs the operation as circuit-breaker and is coupled, therefore, with the gear of the breaker. The additional valve for the elimination of short-circuits, enclosed in the figure by a chain dotted line, is operated by piston (9), which is actuated by the auxiliary valve (8). This valve is opened by the magnet (10), which is energised by the overcurrent relay. Releasing piston (11) interrupts the coupling of the magnet with the auxiliary valve

(8), so that the valve for " short-circuit elimination " closes. The duration of the opening of this valve depends on the rate of the increase of pressure on piston (11), which can be adjusted by a throttle screw. The regular function of the breaker is not influenced in any way by the insertion of the parts within the chain dotted line.

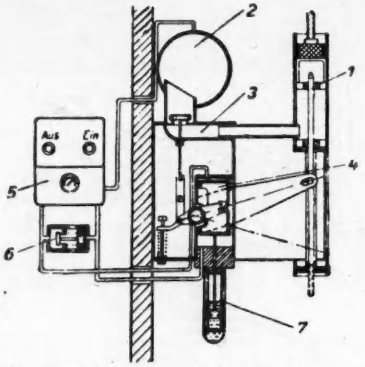


Fig. 2. Wall-mounted air-blast breaker.

1. Nozzle
2. Compressed air tank
3. Compressed air pipe
4. Gear of the breaker
5. Valve
6. Special outlet valve
7. Puffer piston

The wall-mounted type of air-blast circuit breakers for indoor service and voltages up to 30 kV. (Fig. 2) have such a short opening and closing time, that for producing short interruptions the breaker may perform complete opening and closing operations. Nothing has to be changed in the construction of the breaker, only a special valve (6) has to be added, which provides a quick outlet of the air out of the space above the piston, so that the movement of the piston and the rod is not held up during reclosing. The duration of the interruption of this type of breakers can be adjusted by the choice and the setting of the relays, whereas with the open-jet breakers, alteration of the duration of the interruption is performed by influencing the pneumatic process.

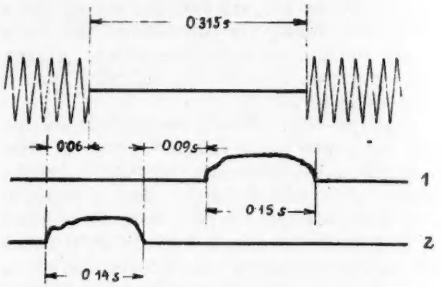


Fig. 3. Oscillogram of an interruption of a wall-mounted air-blast breaker.

1. Current of magnet controlling closing operation
2. Current of magnet controlling opening operation

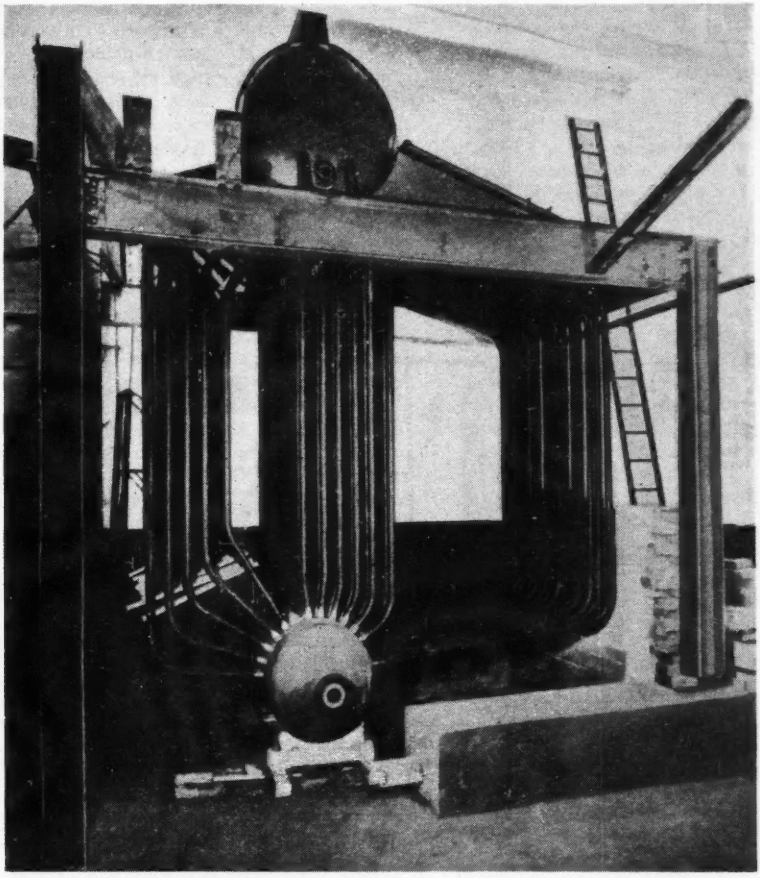
Fig. 3 shows an oscillogram of a 0.315s—interruption of a wall-mounted breaker and of the currents of the magnets, which control the valves. It has been found that the most suitable duration of the interruption is from 0.2 to 0.35s. With shorter times restriking of the arc may occur, whereas customers may object to longer interruptions. With the jet-type breakers, one may achieve even interruptions down to 0.05s, with the wall-mounted breakers the minimum is about 0.2 to 0.28s, on account of the masses, which have to be accelerated.

PROCESSING WITH DOWTHERM

(From *Heat Engineer*, Vol. 14, June, 1939 & Vol. 15, January, 1940).

STATISTICS compiled from more than 30 installations of Dowtherm boilers and heating systems clearly indicate that the use of Dowtherm is the ideal method to obtain processing temperatures above 400° F.

Steam is an excellent medium up to 400° F., but at higher temperatures, the steam pressure becomes so excessive that many difficulties arise. The eutectic mixture of Diphenyl and Diphenuloxide distributed by the Dow Chemical Company under the name of Dowtherm " A," has excellent properties for the transmission of heat between 400° F. and 700° F. Its boiling point at atmospheric pressure is 500° F. and it does not solidify until the temperature is lowered to 54° F. Some installations of the Dowtherm heating system will be mentioned to indicate the simplicity of the method.



A standard Dowtherm vapor generator of 3,000,000 Btu. per hour capacity during construction

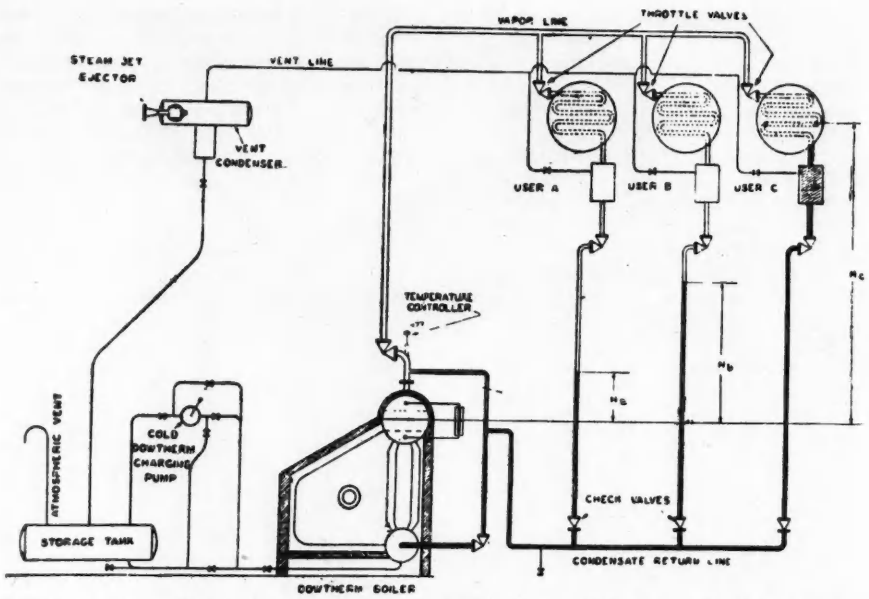
A number of Dowtherm boilers have been installed for heating vegetable oil in connection with continuous deodorization systems. In these processes, it is necessary to heat the oil to 450° F., and if steam were used it would have to be generated at a pressure of 600 lbs./sq.in. The same temperature is obtained with Dowtherm at a pressure of only 12 lbs./sq.in.

The Dowtherm boiler, operating at a gauge pressure of 12 lbs/sq.in., generates vapour at 550° F. The vapour flows to the shell side of a shell-and-tube type of heat exchanger in which it condenses, giving up heat to the oil passing through the tubes. The condensed Dowtherm returns by gravity to the boiler.

Where the necessary difference in height between the heating vessel and the boiler cannot be obtained, the condensed Dowtherm may be collected in a receiver and pumped back to the boiler.

Dowtherm boilers have been installed for heating the oil in batch systems for deodorization of vegetable oil. In the batch process, the deodorizating tank is charged with oil and the heating coils are immersed. Dowtherm vapour is introduced into the coils and the oil is heated under a high vacuum to about 400° F.

A Dowtherm boiler was installed in a plant for heating asphalt for coating of cables. In this case, the asphalt is heated only to about



Gravity return, Dowtherm indirect-heating system having multiple users. The temperature within the user as well as the proportion of active heating surface may be varied by throttling.

400° F., so that the entire Dowtherm heating system is for part of the time under a vacuum. In another plant, Dowtherm is being used to heat asphalt for the purpose of coating wall board.

It is probable that Dowtherm will eventually be used quite extensively in the petroleum refining industry, especially where very accurate temperature control is required, within an operating range of 400 to 700° F. The accurate temperature control is required for sharp fractions in the production of special naphtha and other products having a narrow range of boiling points. The admission of Dowtherm vapour to the various users (kettles, etc.), and the operation of the vapour generator are easily regulated or made automatic.

FACTORS AFFECTING PRODUCTION OF BLAST FURNACE GAS

By J. S. FULTON. From *Iron and Steel Engineer*, Vol. 17, No. 9, September, 1940, pp. 68-69).

THE iron blast furnace in this paper will be analysed on the basis of the character of its top gas. If it were to act solely as a gas producer, to produce gas for commercial combustion elsewhere, then little or no CO_2 would be generated. Since Fe_2O_3 must be reduced in order to make pig iron, the amount of O_2 transferred from the ore to the CO determines the amount of CO_2 formed, and so alters the quality of the gas.

Some blast furnace operators feel that it is a reflection on the blast furnace to consider it as a gas producer " because the furnace is run to make pig iron." In so doing, they do not differentiate between the primary purpose and the primary function of the furnace. The primary function of the furnace, once the fire has been lit in the hearth, is the production of gas, and unless the gas has the proper percentage of CO, no iron can be reduced, and so it

defeats the primary purpose of the operation. The weight of top gas is several times that of the pig iron because it includes all of the weight of the blast, oxygen in the ore, CO_2 calcined from the limestone, nearly all the carbon and all of the moisture entering the furnace. The weight of the blast is at least double that of the iron produced.

Since the top gas reports the final combination between carbon and oxygen, it has balanced the carbon in the coke against the oxygen received in any manner, i.e. O_2 in blast, O_2 in ore, O_2 in CO_2 from limestone, O_2 from H_2O dissociated anywhere in the furnace. By a coincidence, the C in CO_2 calcined from the limestone about equals the weight of C needed for the pig iron and flue dust.

This relation may be set up as a mathematical equation, and neither side of this equation may be changed without making a compensation on the other side. If the O_2 in the blast or in the solids is increased, it is necessary for a relative amount of carbon to shift from CO into CO_2 if the carbon rate does not change. This increase in CO_2 decreases the CO/CO_2 ratio from that previously shown.

Little attention has been given to the influence of the amount of hydrogen shown in the top gas other than as a cause of a minor change in the percentages of N_2, CO and CO_2 and a more marked change in the Btu value for further combustion. The Brassert curves show about $3\frac{1}{2}\%$ by volume of H_2, which represents about 20 lb. of H_2 in the lower carbon rates and was accompanied by 160 lb. of O_2 that had to be balanced in the equation A. This H_2 comes both from the blast and also in at least equal amounts from the dissociation of water accompanying the solids into the furnace. If the furnace continues to use the same carbon rate with an advance of H_2, it automatically increases the CO_2 and so decreases the CO/CO_2 ratio.

The writer suggests the desirability of the blast furnace profession getting together and preparing new curves of top gas production. In order to provide a better basis of comparison, at least three sets are necessary, distinguished by the pounds of hydrogen per long ton as 20, 30 and 40 lb. Each one of these will show a different CO/CO_2 ratio for the same pounds of carbon per ton. Since some of the corporations are basing their production on 2,000 lb. tons, it will be necessary to develop another set of curves for this ton.

Before these curves are developed, it will be advisable to recognise that the present basis, "cubic feet of standard air 60°F., 29.92 in Hg.," gives no consideration to moisture in the atmospheric air and the amount of oxygen in this moisture.

Any change in the weight of C and O_2 supplied per ton to the furnace from any source must make a change in the top gas analysis, even though momentarily the furnace may ignore the rate of supply and over eat or diet as it sees fit.

APPLICATION OF DEMAGNETISING AND IR DROP COMPENSATION TO COLD STRIP MILLS

By G. E. Stolz. (From *Iron and Steel Engineer*, Vol. 17, September, 1940, No. 9).

While the steel industry has produced strip either in single stands or in tandem mills for a period almost as long as we have had a steel industry, more advance in the art of producing strip has been made in the last six years than during the entire previous period. Previous to this time, it was seldom that strip was rolled at a speed in excess of 300 ft. per min., but since that time there has been a continual increase in speed at which strip is rolled until to-day tinplate is being produced in a tandem mill in excess of 2,000 ft. per min.

These mills are not only operating at higher speeds, but considerable improvement has been made in the finish of the sheet and the uniformity of gauge. With this rapid change in practice, it has become necessary for the electrical manufacturers to provide driving equipment which will make it possible to operate these mills satisfactorily at higher speeds and obtain uniform gauge without causing too many breaks.

The finished gauge of the strip is the result of the combined roll pressure on individual

stands and tension between stands. Roll pressure depends upon the setting of the screws while tension between stands is fixed by the speed adjustment of the driving motors.

As higher speeds are used, it becomes more important to provide motors that have good speed regulation so as to maintain uniform gauge and avoid tearing the steel in two.

Recently, electrical equipment has been furnished to drive a tandem tinplate mill which has been provided with auxiliary apparatus that compensates for the demagnetising effect of the armature on the main field and the IR drop in the armature circuit.

This mill was started in April, 1940, and the compensating equipment which has been provided with the motors has demonstrated from the start of this mill that it constitutes a distinct improvement in the design of electrical equipment for a mill of this type.

MILLION-POUND PER HOUR BOILER TOPS SHERMAN CREEK STATION

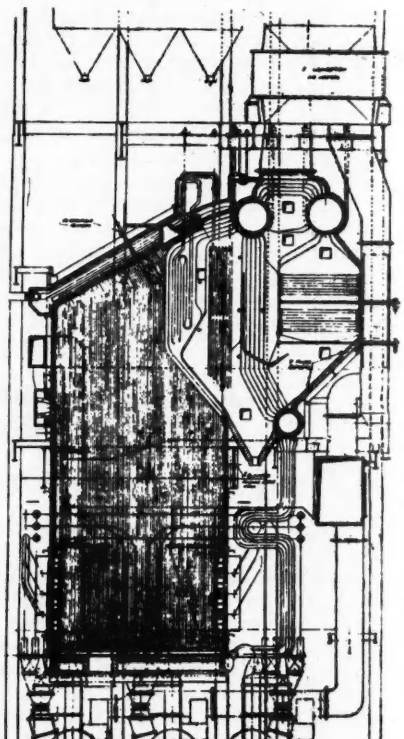
(From *Combustion*, Vol. 12, No. 3, September, 1940, p. 34).

THE latest extension in the construction programme of the Consolidated Edison Company of New York, Inc., is the topping of the Sherman Creek Station, located at 201st Street and the Harlem River. This 200-lb. station, placed in operation in 1913, and increased in capacity from time to time, contains six turbine-generators aggregating approximately 140,000 kw.

This topping installation will consist of a million-pound per hour steam generating unit, designed for 1,775 lb. pressure and supplying steam at 1,600 lb., 955° F. to a 50,000 kw., 3,600 r.p.m. turbine-generator exhausting to the present 200 lb. station heater.

The steam generating unit will be of the three-drum bent-tube type with Elesco superheater and economiser, two Ljungstrom regenerative air preheaters and a steam washer in the steam drum. In addition to the twelve coal burners, three in each corner, there will be two oil burners in each corner, and four horizontal oil burners located in the front upper wall.

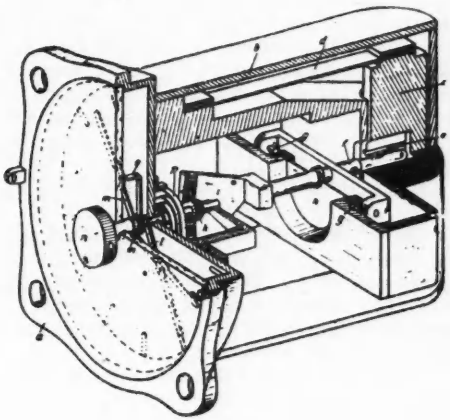
Plain bifurcated tubes will be employed for the front, rear and side furnace walls, and the roof and bottom will have fin tubes. The unit itself will occupy a space of 49×53 ft. between building columns and will extend approximately 90 ft. to the top of the air heaters, the mills being located at the basement level.



THE "DVL" ACCELEROMETER

By H. FREISE. (From *Luftwissen*, No. 2, February, 1940).

ACCELERATIONS taking place at the centre of gravity of an airplane are indicative of the loads imposed upon the structure. The greatest loads are encountered during pull-outs from drives or tight turns during manœuvres. The knowledge of these accelerations is important in test flying, medical flight research, atmospheric turbulence investigation, as well as in regular military flying. There are several types of accelerometers and acceleration recorders. The instrument described in the article is a compact dashboard type unit whose whole purpose is that of an indicator. The accompanying sketches show the principle of its operation and the mechanism of the actual unit. Mass (C), supported by two flat springs (D), undergoes vertical displacement which is a function of the acceleration imposed on the airplane. In the cross-sectional view of the instrument are shown the essential constructional details and their position.



"ALL-ANGLE" MILLING MACHINE.

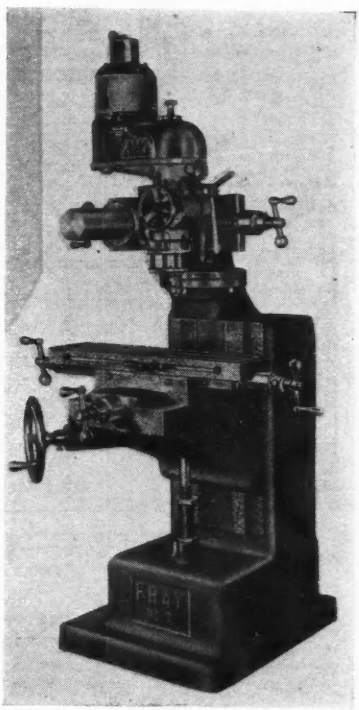
THE illustration brings out clearly the essential features of construction and operation of the No. 7 "All-Angle" milling machine, designed and manufactured by the Fray Machine Tool Co., Glendale, California.

The column of this milling machine is made of Meehanite G.C. metal, as are also the knee, saddle, table, turret head and so on. These castings are normalised. The column is heavily cross-ribbed on the inside for rigidity. The base is formed pan-shaped in order to catch chips, oil or cutting coolants.

The feed nuts are made of Ampco metal, and are adjustable for wear. All dovetailed slides are properly gibbed for adjustment. Positive locks are provided for knee, saddle, table and turret cross slide.

The table is provided with three standard T-slots for clamping work and fixtures. All feed screws are fitted with large bevel-cut dials with graduations in .001 in. The feed screws are mounted in Ampco metal bushing and are fitted with precision ball thrust bearings, with adjustment for wear.

The turret head rotates a full 360°, and is locked at any point by four $\frac{3}{4}$ in. bolts. Cross travel above the turret converts the machine into a full universal miller.



"LOW-HEAD" COOLING TOWERS

(From *Heat Engineer*, Vol. 13, November, 1938, No. 11 & Vol. 15, April, 1940, No. 4).

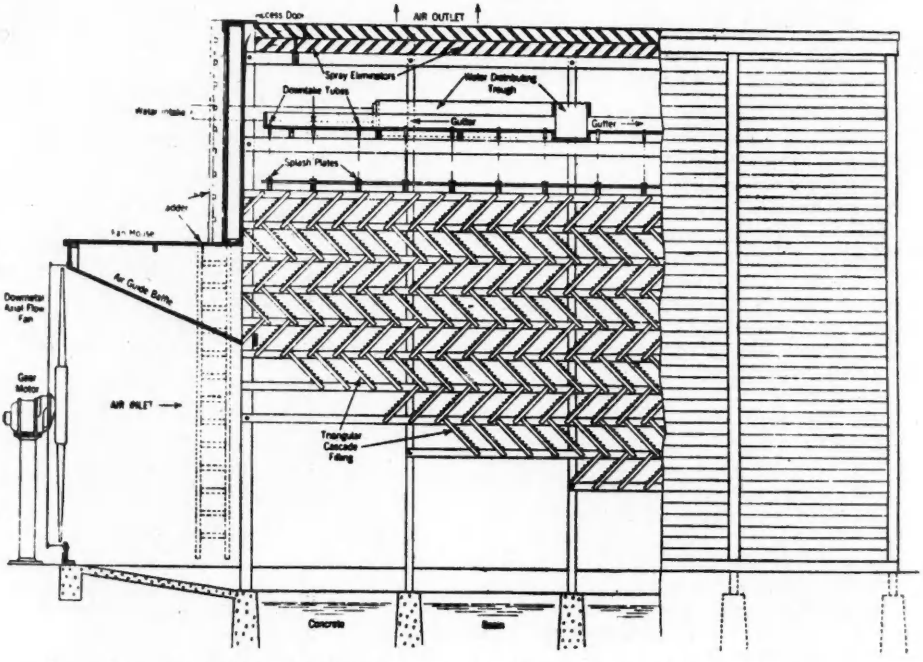
THE largest forced draft cooling tower of the modern Low-Head type has been in service at the Arsenal Hill Station of the Southwestern Gas & Electric Company at Shreveport for the last year. The tower consists of two sections, forming one continuous structure, having 20 individual forced-draught cells, each powered by an axial flow fan 12 ft. in diameter. The total capacity is 43,000 gallons of water per minute, cooling it from 97.6° F. to 86.75° F.

One of the major factors in the operating cost is the head against which the circulating pump must work. The filled tower overhead gravity distribution system has an inherently lower pumping resistance than the atomizing nozzles used in spray towers.

In addition, the fact that the operating cost of the low-head tower is less than that of the standard type of forced draught tower, the initial cost is less due to the lower overall height and smaller area subject to wind load. The result is a simpler frame design.

Although the main difference between the low-head tower and the standard tower is the lower pumping head of the first one, other improvements have also been made. The fans are placed further away from the falling water in the plenum chamber at the base of the tower. Air is discharged into a diffusion chamber, the roof of which slopes downward from the top of the fan towards the tower interior.

An even distribution of the air over the entire width of the cell is effected in the diffusion chamber, and losses due to turbulence



Cross section of forced draft cooling tower showing arrangement of fans and filling resulting in a low pumping head.

are minimized. This improves the efficiency of conversion from high air velocity, leaving the fan to the low pressure required to force the air upward from the plenum chamber through the staggered rows of filling, and thence, to the atmosphere.

The distance of the fans from the filling and the downward slope of the air guide baffle, forming the roof of the diffusion chamber prevent the fan blades from moisture.

The advantages of the low-head towers may be summarized as follows:

(1) Low operating cost due to low pumping head.

(2) Low initial cost due to reduced height.

(3) Efficient conversion of air velocity into low pressure.

(4) Wetting of fan blades is prevented.

GERMAN MACHINE TOOL ACCESSORIES

(From *Die Werkzeugmaschine*, Vol. 44, No. 2, January, 1940, pp. 29-35).

Clamping chucks for thin work pieces.

DRP. 662.898. The overall diameter of this clamping device can be kept at small values, and allows quick inserting and removing of the work piece by a simple handgrip. It consists of a main part (1), Fig. 1, into which the work piece is introduced. The few remaining parts are built into the main part. In a prismatic slot of the main part, a double armed lever (2) is fixed, which can rotate round the axis (9). In the front end of the lever, a short hole is drilled (3), which is slightly inclined to the axis of the tool when in working position. The end of the work piece penetrates this hole. The back of the lever arm has a chamfered surface (4) against which a cam (5) with a correspondingly chamfered end is pressed by the action of a spring. The work piece is rigidly fixed in the clamping chuck by slightly bending the end of it. This bending is kept within the range of elasticity. The work piece can easily be removed by pushing back a slider connected to the cam (5), hence, the pressure of the cam on the lever is discontinued, and the work piece can be withdrawn. A stop (8) in the main part of the chuck limits the amount of bending of the work piece.

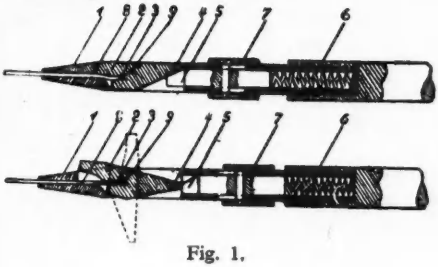


Fig. 1.

Clamping chucks for machine tools.

DRP. 663.222 (André Bechler). Fig. 2a shows the front end of the head stock spindle of a lathe in which an internally conical bearing surface (b) holds the head (c) of a clamping chuck. The back of the clamping chuck is denoted by (d). The back end (d) is provided with an angular thread (e) engaging the internal

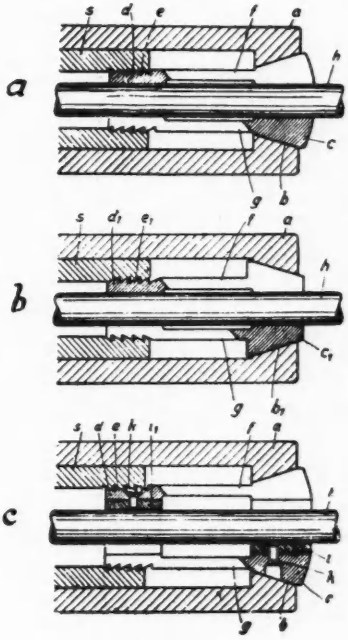


Fig. 2.

thread of a hollow shaft (s). The clamping chuck is slotted at (f) on the front end and at (g) at the back end. The work piece is fixed into the axis of the head stock spindle.

If a pull is exerted on the shaft (s), so as to pull the front end (c) into the cone (b), a perpendicular pressure is transmitted through the threads (e), pressing the clamping chuck firmly on (h). Thus, the internal diameter of the front of the clamping chuck is the same as that at the back end, so that the work piece is gripped equally at both points.

In Fig. 2b, the conical end of the clamping chuck is reversed. Also, the threads (e) are cut in the reversed direction. Hence, clamping of the work piece is obtained when the shaft (s) is pushed forward instead of being pulled. Otherwise the arrangement and operation of the device is identical with that shown in 2a.

In case illustrated in 2c, the general arrangement is the same as in 2a, but in addition the clamping chuck is fitted with lining (i and i') which are kept in position by small screws (k) and can be easily changed.

Clamping chucks with radially moving jaws.

The main patent 630.720 refers to a clamping device with radially moving jaws operated by a sleeve, and secures the sleeve in the required position by means of spring controlled wedges. The latter are fixed between the jaws tangentially along the circumference of the lining. In some cases, however, there is not enough room between the jaws, and in this case the invention DRP. 666.614 (J. Scheelen) is very helpful.

The body (1), Fig. 3, is fixed to the spindle of the lathe, and carries three radially movable jaws (2). They are moved by the sleeve (3) which can slide over the jaws and has a stroke (h). The movement of the sleeve in the horizontal direction causes the jaws to move radially as f.i. by means of a dovetail shaped guiding groove. A small plunger is placed behind each jaw in the lining. These plungers are forced outward by springs (5). The ends of the plungers press against the conical surface of the sleeve. The cone is diverging in the opposite direction to that on the front of the clamping chuck. Hence, if the sleeve is moved to the left, the jaws are gripping on the work piece, and the plungers (4) are allowed to move radially outward at the same time. They thus prevent the sleeve sliding back due to vibration, etc. To increase the resisting force of these wedges they are provided with surfaces inclined to their axis as shown in Fig. 3a.

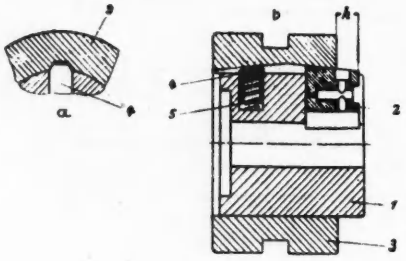


Fig. 3.

Clamping device with spring controlled membranes.

The invention of A. G. F. Aallgren (Aktiebolaget Nomy) DRP. 661.772 overcomes the difficulty of ordinary clamping devices of this type, which might loosen the grip under the action of centrifugal forces if the machine is running at high speeds. The clutching disc

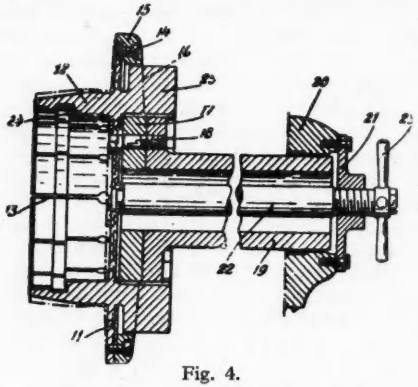
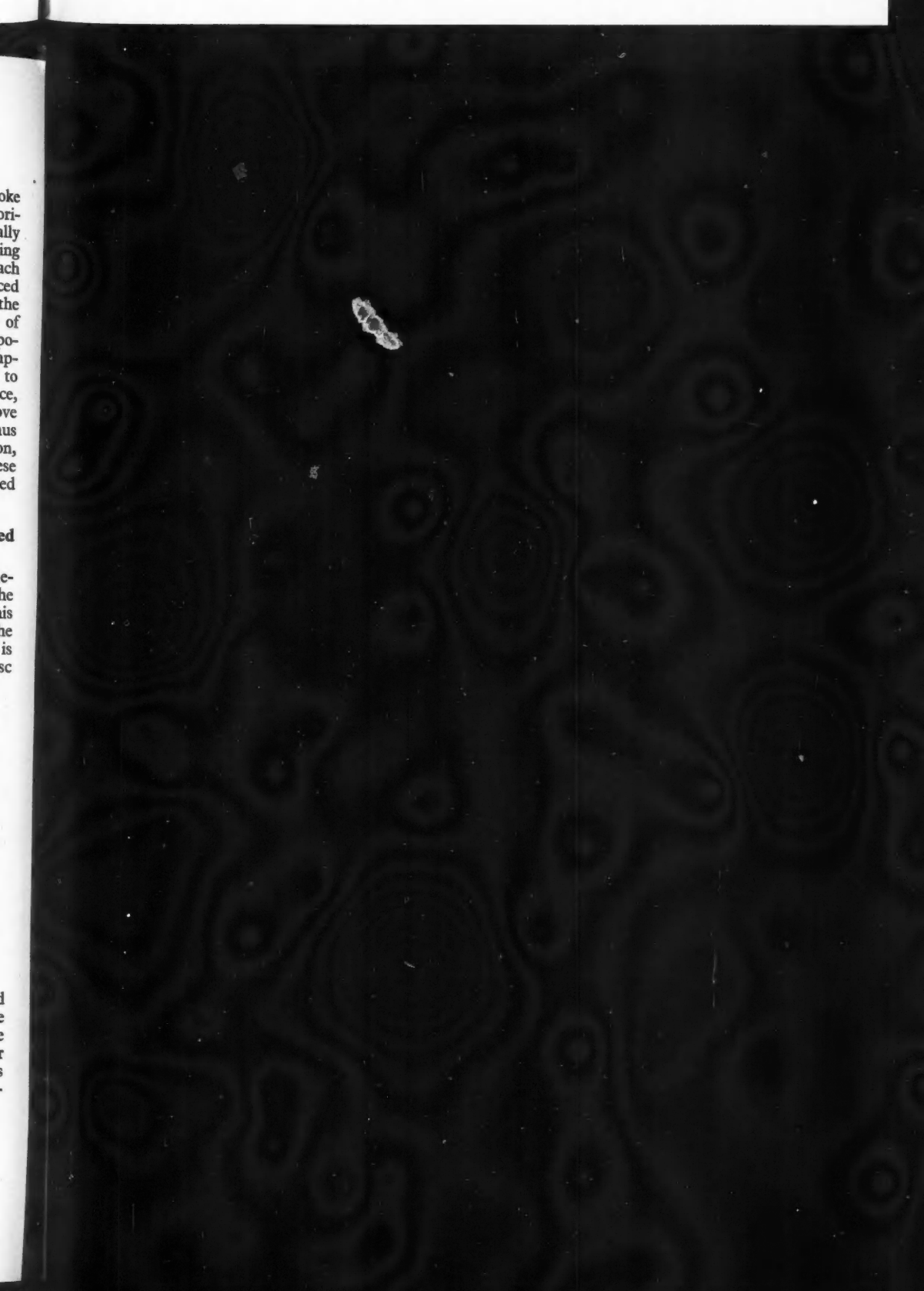


Fig. 4.

(11), Fig. 4, has paw-like parts (12) separated from each other by slots (13) and cast in one with (11). The rim of (11) is screwed to the rim of wheel (15). The hub of the latter (17) is fixed to the sleeve by means of screws (18). The sleeve (19) rotates in bearing surfaces (20).

(To be continued).